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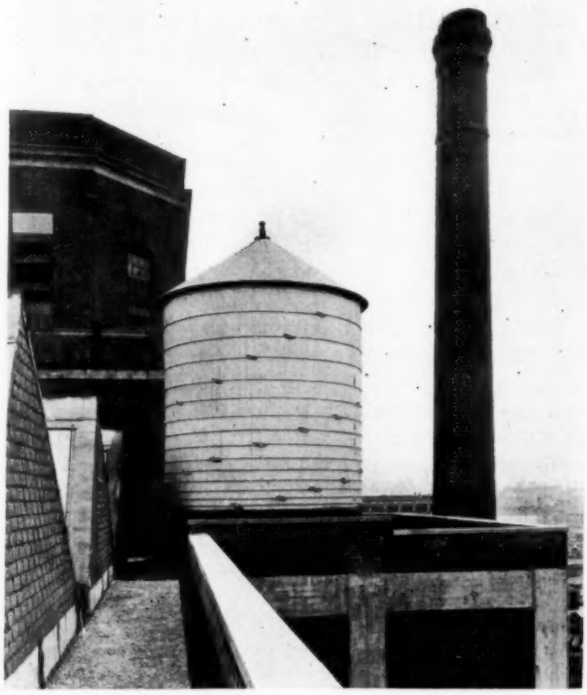
Bayuk Cigars Inc. gets

By R. D. Touton*

Increased Air Conditioning Capacity *with* Economy

by means of Off-Peak Compressor Operation



The 25,000 gallon tank which stores cooled water for the "off-peak" air conditioning system. The pent house with open windows is the tenth floor enclosed spray pond for cooling condenser water for 120-ton compressor

AN alteration in manufacturing operations at the central plant of Bayuk Cigars Inc., Philadelphia, imposed an additional load on the air conditioning which required an increase in capacity. This plant is completely air conditioned; twenty-six individual systems—many of which include the use of refrigeration during all or part of the year—are operated continuously to establish proper atmospheric conditions for all stages of tobacco processing and the manufacture of Bayuk "Phillies" and Prince Hamlet cigars. Although practice has been to incorporate adequate conditioning equipment as an integral part of original plant design, certain modifications and additions become necessary from time to time in order that advantage may be taken of new methods and because of increased production.

It is the purpose of this article to describe how the problem of providing for the increased air conditioning load was solved by the "off-peak" storage of refrigerating effect from the existing 120-ton centrifugal refrigeration compressor. The required refrigerating effect is obtained by off-peak operation of the compressor at a total energy cost of less than one cent per ton hour of refrigeration. Considering that it is of course unnecessary to include any overhead charge on the primary refrigeration equipment against the off-peak system, the operation is remarkably economical.

Production Changes Add to Air Conditioning Load

The production changes which added to the air conditioning load included the use of Cellophane as a protective wrapping material, and the adoption of a standardized artificial daylighting system to insure uniform product inspection during all weather variations, day and night. The 40 Cellophaning machines form an individual case for each cigar by fusing together the edges and ends of the wrapping material, accomplished by moving a constantly heated 300 watt electric plate into position for brief contact with the wrapped product. In the selecting department, sixty 500-watt color correcting lighting units are employed to maintain the desired illumination.

These additional sources of heat total 42 kw (2390 Btu per minute) and exist during the cigar production hours,

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7 a. m. to 10 p. m. Naturally, such an increase in the load on the air conditioning plant could not be handled by the original system. An important consideration in the alteration of the air conditioning system was the tendency of the Cellophane to become tacky when exposed to temperatures much above 80 F.

Offices Require Refrigeration

Wrapping and inspection are conducted on the 8th or top floor of the building, and the general and private offices occupy the next lower floor. In the original design, these areas were provided with an independent fan system, including a filtered fresh air intake, heating coils and means for introducing treated air from the central station humidifier as required to maintain the proper humidity during the dry winter months. This same arrangement was employed to secure cooling and dehumidifying effect for the offices during the summer, when the central unit was in operation as a dehumidifier.

With this refrigeration in operation, a reasonably pleasant and satisfactory condition was obtained for both the offices and packing department. However, as tobacco is best made into cigars at a relative humidity of approximately 70 per cent, the central station units (one in the basement serving the 1st to 4th floors inclusive and another in a 9th floor penthouse serving the 5th to 8th floors) are not used as dehumidifiers until the outside wet bulb temperature reaches 69 F. Therefore, there was a considerable amount of milder but somewhat uncomfortable weather in the spring and fall when further cooling was desirable for the 7th and 8th floors. This situation was accentuated by the fact that elevators and stairwells discharged an appreciable amount of higher dew point air into the two upper floors from the manufacturing departments—a desirable condition during the winter but one which created some discomfort during the warm months.

Well-Insulated Building, Available Ducts, Simplify Job

Due to the 2 in. cork insulation and double sash originally installed in the building, the load under consideration was somewhat lower than would otherwise be expected. Using the conventional factors, a total of 15 tons was indicated for the packing department through both manufacturing periods, or a total of 15 hours per day, and a load of 10 tons of 9 hours daily duration for the offices.

Duct systems and individual fans were available on each floor, so that the situation presented an excellent opportunity for the placing of coils in the path of the air flow. Accordingly, the surface type of cooling and dehumidification was decided upon.

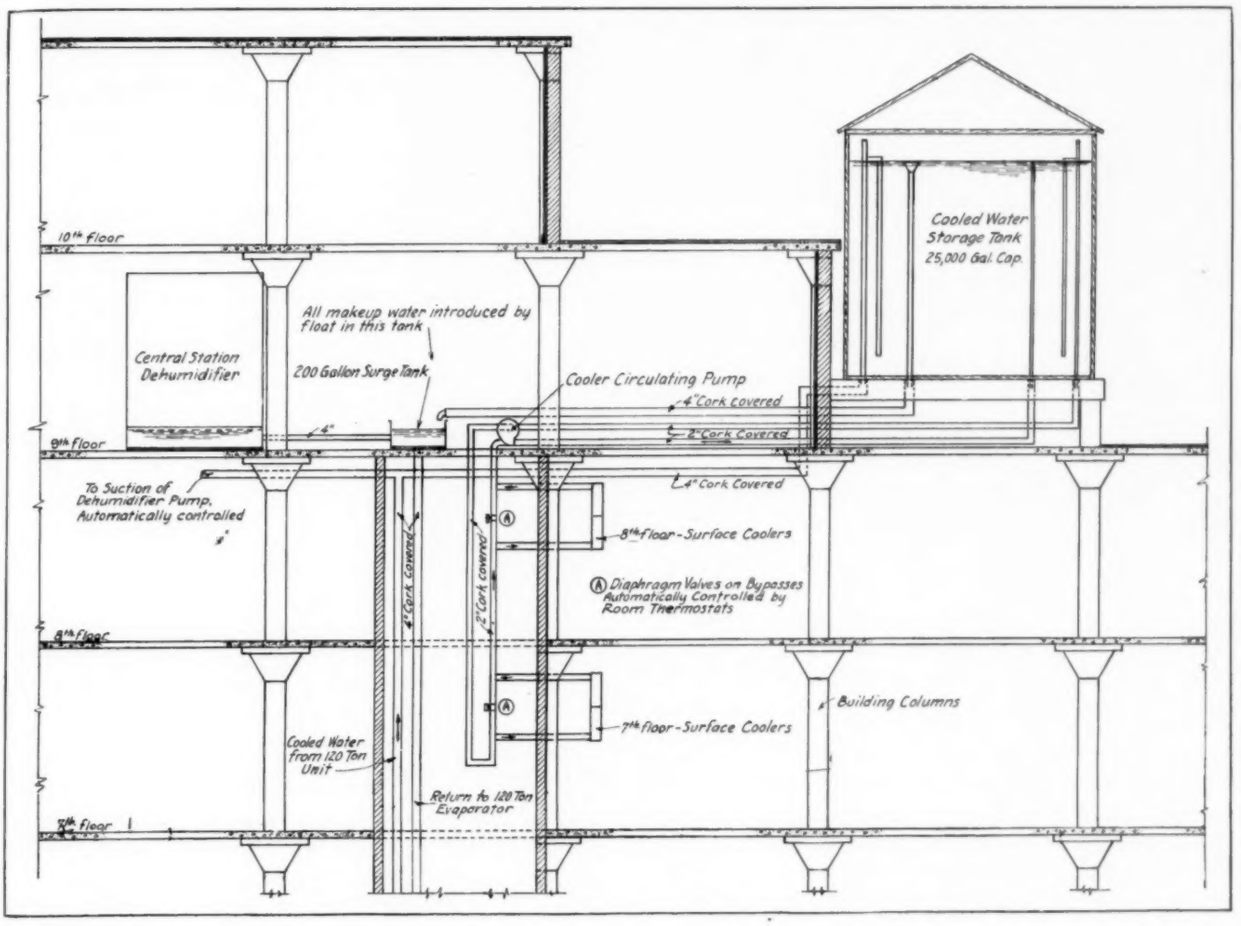
Compressor and Off-Peak Operation Studied

Plans for installing the required 25 tons of refrigeration capacity were then considered. Serious difficulty was found in securing a location for an additional compressor and evaporator. The basement presented the only logical site, but developed and proposed process work for improving tobacco treating methods had utilized or "earmarked" the extra space provided when the building was designed and constructed. Furthermore, such a location would necessarily require long insulated lines for carrying the cooled water to the points of use; over 400 ft of such piping would have been needed and would have crowded the original pipe shaft, making the lines inaccessible for maintenance.

As mentioned, a solution without these limitations and with many other advantages was found by arranging to accumulate the cooling effect from a few hours' operation of the existing 120-ton centrifugal compressor installation. Storage for the cool water is provided in a tank located close to the cooled and dehumidified areas.

A complete analysis was made of the four year experience in the operation of this compressor. Its capacity of 120 tons when operating in the 70 to 60 F range was found to agree closely with the maximum requirements, and was nicely adequate to insure comfortable working conditions and avoid the usual too tightly packed cigars during humid weather. This made it inadvisable to add an additional load during the hours of regular operation without seriously impairing this satisfactory treatment for the cigars and operators in other depart-

RARELY IS AN air conditioning system installed in a plant or building and then forgotten except for ordinary operation and maintenance. Load requirements change, it is desired to condition other departments or areas, or a new process or department may be added..... At this Bayuk Cigar plant (for instance) an alteration in manufacturing operations and the need for air conditioning office space increased the air conditioning load. Additional capacity was provided by arranging for the storage of refrigerating effect (cooled water) from the existing compressor..... While refrigeration is obtained very economically, the most important result is that the new methods of cigar inspection and wrapping have been provided with the same high standard of air conditioning that Bayuk has found essential in insuring smoking quality and accurate production control.


10th Floor
Cooled Water Storage Tank 25,000 Gal. Cap.
All makeup water introduced by float in this tank
Central Station Dehumidifier
200 Gallon Surge Tank
Cooler Circulating Pump
4" Cork covered
2" Cork Covered
9th Floor
4"
To Suction of Dehumidifier Pump. Automatically controlled
4" Cork Covered
8th floor - Surface Coolers
Diaphragm Valves on Bypasses Automatically Controlled by Room Thermostats
8th Floor
4" Cork covered
2" Cork covered
Building Columns
7th floor - Surface Coolers
Cooled Water from 120 Ton Unit
Return to 120 Ton Evaporator
7th Floor

Layout of the 320 ton-hour off-peak-refrigeration air conditioning system at the Bayuk plant

ments. It was also true that, when operating under these maximum conditions, the temperature ranges of the cooling medium were such as to preclude use of this water for a satisfactory office conditioning system.

However, such peaks as these were found to exist for only relatively short periods of consecutive hours. The tabulation below indicates the duration of these periods during the 1931 season (immediately preceding the installation of this off-peak system) in which the total number of hours of operation was considerably higher than in any of the three previous years, and was the first season in which it was necessary to continue compressor operation into the night hours:

Period No. 1	9 hours
Period No. 2	63 hours
Period No. 3	5 hours
Period No. 4	6 hours
Period No. 5	10 hours
Period No. 6	18 hours
Period No. 7	8 hours
Period No. 8	9 hours
Period No. 9	15 hours
Total full capacity hours	143

In all cases except No. 2 these peak demands were on non-consecutive days.

The new loads, for which it was desirable to supply a maximum of 25 tons refrigerating effect in the 40 to 60 F range, could be expected to exist at least in part for a total of 2000 hours during the season.

Since these sources of heat were caused primarily by manufacturing processes, sun effect, office occupancy, lights, motors, etc., rather than by weather conditions, the 25 ton capacity would be required only during office hours, and the operation would continue until the close of the day's manufacturing with a maximum of 15 tons. Incidentally, this latter part of the daily schedule would be serving a department requiring a somewhat higher dewpoint than the office. The total net duty thus arrived at was 320 ton hours.

It was evident, since the existing compressor could be calculated to perform this service in approximately 5 hours operation in each 24 (taking into consideration the reduced output at lower evaporator temperatures), that provision of an ample storage means would satisfy the problem.

With this established range of 40 to 60 F a total effect of 166 Btu would be available for each gallon of water cooled. The net requirement of 320 ton hours plus a provision of 10 per cent for losses, equalled approximately 350 ton hours or 4,200,000 Btu.

Several storage arrangements were considered: At the outset it was thought advisable to locate the required

storage, completely insulated, at some point in the penthouses in order to avoid sun effect on the cooled water. Again available space prohibited such an installation and after investigation decision was made to locate the tank on the roof. Additional calculations seem to minimize the necessity for any insulation other than that provided by the wood itself. Considering a tank of 2¾ in. wood with a double roof, the result (on the basis of:

(1) A sun effect of 8 Btu per sq ft per hr on one-half of the shell area.
(2) An outside temperature of 95 F.
(3) An average water temperature of 50 F.
(4) A transmission factor K of 0.3 for the wood itself.)

indicated an approximate total loss due to sun effect and transmission of 8 per cent of the total ton hours. Because of the margin of safety in permissible time of compressor operation, and low cost of power during these hours, additional insulation was not included. The tank was painted with two coats of aluminum paint when installed and one each year since. It is a round cypress tank of 25,000 gallon capacity, with both flat cover and conical roof, located in a spot where the penthouses provide as much shading effect as possible (about 25%).

The primary piping system is so arranged that during periods of light loads on the central dehumidifiers, the excess cooling effect can be diverted to the tank, effecting a reduction in the number of hours of night operation required for cooling the body of water for the next day's operation. This method created a noticeable improvement in the flexibility of control for the large compressor during such small tonnage demands.

Provision was made for completely separating the two systems. Cool water is delivered to the bottom of the tank and return water for the evaporator taken from the upper level. This return line is connected to an open surge tank at the level of the 9th floor dehumidifier, permitting partial return of the circulating medium from the two systems coincidentally. The total amount of water pumped is 220 gpm. There is necessarily a lag in the water flow returning, dependent on the head which the water in the tank must develop above the return opening to establish the required flow. Study of weir effects indicated that the 4 in. return line should be provided with a bell mouth 14 in. in diameter at the water level. With this arrangement a total rise of approximately 1 in. is required to attain the full return, and the total number of gallons required from the surge tank to start the flow, and consequently the number which the same tank must receive when operation ceases, is reduced to such a point that a relatively small surge tank can provide this capacity without an undue change in the humidifier water level.

Sufficient cooling surface was installed in the two duct systems in such manner as to carry the loads with a maximum water temperature of 60 F. The water from the bottom of the tank is delivered through a single circulating system to the 7th floor offices first, then through the packing department coolers on the 8th floor, finally returning the doubly warmed water to the top of the tank. As mentioned, the office (lower dewpoint) demand terminates at 5 to 6 p. m., leaving only the higher temperature condition to be cared for during that portion of the schedule when the tank temperature is at its highest.

Control is provided by means of automatic bypass valves for each system.

Readings Show Tank Temperatures

Readings taken of the tank temperatures on a warm bright Sunday, when no part of the new load existed, confirmed the design allowances for transmission and sun effect. With an average tank temperature of 42 F at the start of the day, the rise due to these factors alone on such a day was approximately 4 F.

The temperature of the water at the beginning of the nightly cooling period naturally varies directly with the air temperature during the preceding day, usually being within the 57 to 62 F range. Cooling of the tank water is deferred until the early morning hours, and the surface coolers are used during this water cooling period. In this way, the room temperatures are somewhat reduced before manufacturing and office hours, and the real load is then assumed with a full potential of cooling medium.

This amount of daily compressor operation is carried on in the night hours and occurs during an off-peak period of electrical consumption. Therefore, instead of increasing the billing demand, this energy simply results in an improved electrical load factor, and appears only in the lowest bracket of the wattage charge.

Dunhill's Air Conditioned Humidor Has Capacity of 750,000 Cigars

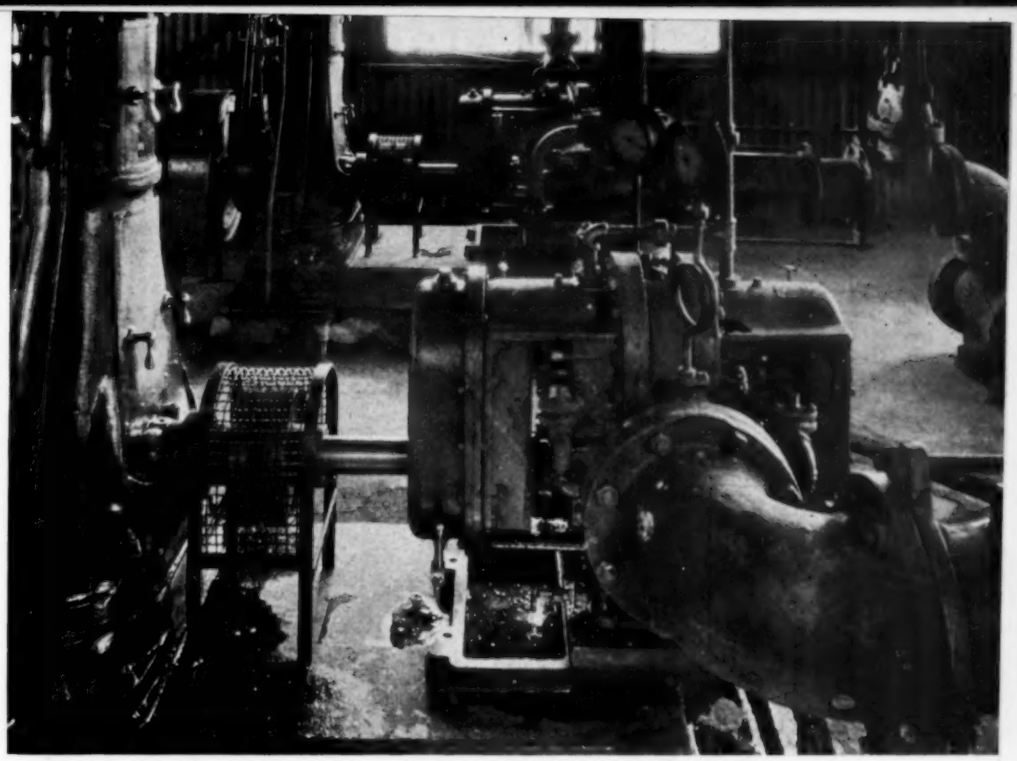
GENERALLY conceded to be the world's finest is the air conditioned cigar humidor on the second floor of the Dunhill shop in the British Empire Building at Rockefeller Center, New York. The architects for the shop were Eugene Schoen & Sons.

The humidor has a storage capacity of 750,000 cigars, and is 50 ft long, 28 ft wide and 10 ft high. It is entirely lined with solid cedar, backed with 2 in. of cork and all of the windows are double glazed. Sixty-four degrees dry bulb and 75 per cent relative humidity are maintained. These conditions are met in part by the steam jet central air conditioning plant described in the May, 1933, HEATING, PIPING AND AIR CONDITIONING. During the winter a higher humidity than that carried by the central system is required, produced by three unit type humidifiers. These units are located within the humidor built into special bins and concealed with cedar louvers. They are fully automatic in operation and maintain the humidity within plus or minus 1 per cent.

The cigars are arranged in the bins so that the conditioned air circulates freely. They are not wrapped, but simply tied together in small bundles of fifteen or so; this method of packing is termed "cabinet packing" or "English market packing."

According to George W. Korell of Dunhill's, cigars kept under these conditions become very pliable and elastic. They make a good, cool smoke and have a fine mellow flavor.

An unusual feature of the humidor is the number of bins in which private stocks of cigars are kept. Many prominent men keep their cigars here. Some of them insist the cigars be conditioned or aged from two to ten years before they will smoke them.—ANDRÈ MERLE.*



Large pump handling hot resin. Steam is circulated through passageways for heating-up purposes, and after operation starts, water is circulated to keep stuffing box temperature at the proper point

Handling Process Materials *that require* Melting

By Lee P. Hynes*

CONSTANTLY increasing use is being made of pipe lines for conveying materials that are too viscous or too solid for practical flow at ordinary temperatures. Materials that require heating before they can be handled by pumping may be roughly divided into two classes: (Class 1), Materials whose viscosity varies according to their temperature over a wide temperature range, such as heavy oils, molasses, tar, and so forth; (Class 2), materials that are solid at ordinary temperatures and that have a narrow temperature range within which they melt and flow freely, such as lard, tallow, paraffin, waxes, resins, fatty acids, shellac, sulphur, pitch, and many special compounds.[1]

Some phases of applying heat to reduce the viscosity of the first class of heavy liquids have been covered by the author in previous articles. This article will deal particularly with the second class of materials and with special types of heating, mixing, and conveying equipment necessary to overcome the special problems presented. Important among these problems are safety, resistance to corrosion, proper distribution and control of the heat input.

Amount of Heat to Supply

In heating any fluid of the first class the heat energy required depends upon the specific heat of the fluid being handled and the temperature rise necessary for sufficiently reducing its viscosity. In heating "Class 2" materials which are normally solid, it is necessary not only to supply heat units to warm the material up to its melting point, but also an additional amount of heat energy to supply the required heat of fusion to melt the material. The heat of fusion may represent the major part of the heating work required and this must not be overlooked in figuring heating capacity for any job. After fusion, further heat will be required to raise the liquid to its desired final temperature.

The following example illustrates solution of a problem of this kind for a batch of 2000 lb of paraffin having an initial temperature of 80 F which is to be melted and raised to a final temperature of 160 F:

Initial temperature of paraffin	80 F
Melting point of paraffin	130 F
Temperature rise required to raise to melting point	50 F
Specific heat of paraffin	0.69
Latent heat of fusion	64 Btu per lb
Temperature rise after melting	30 F

$2{,}000 \times 50 \times 0.69 = 69{,}000$ Btu to raise 50 F
$2{,}000 \times 64 = 128{,}000$ Btu to melt
$2{,}000 \times 30 \times 0.69 = 41{,}400$ Btu to raise 30 F

Total = 238,400 Btu

These figures do not include the relatively small losses due to radiation and to absorption by the metal container. It will be noted that the process of fusion alone requires

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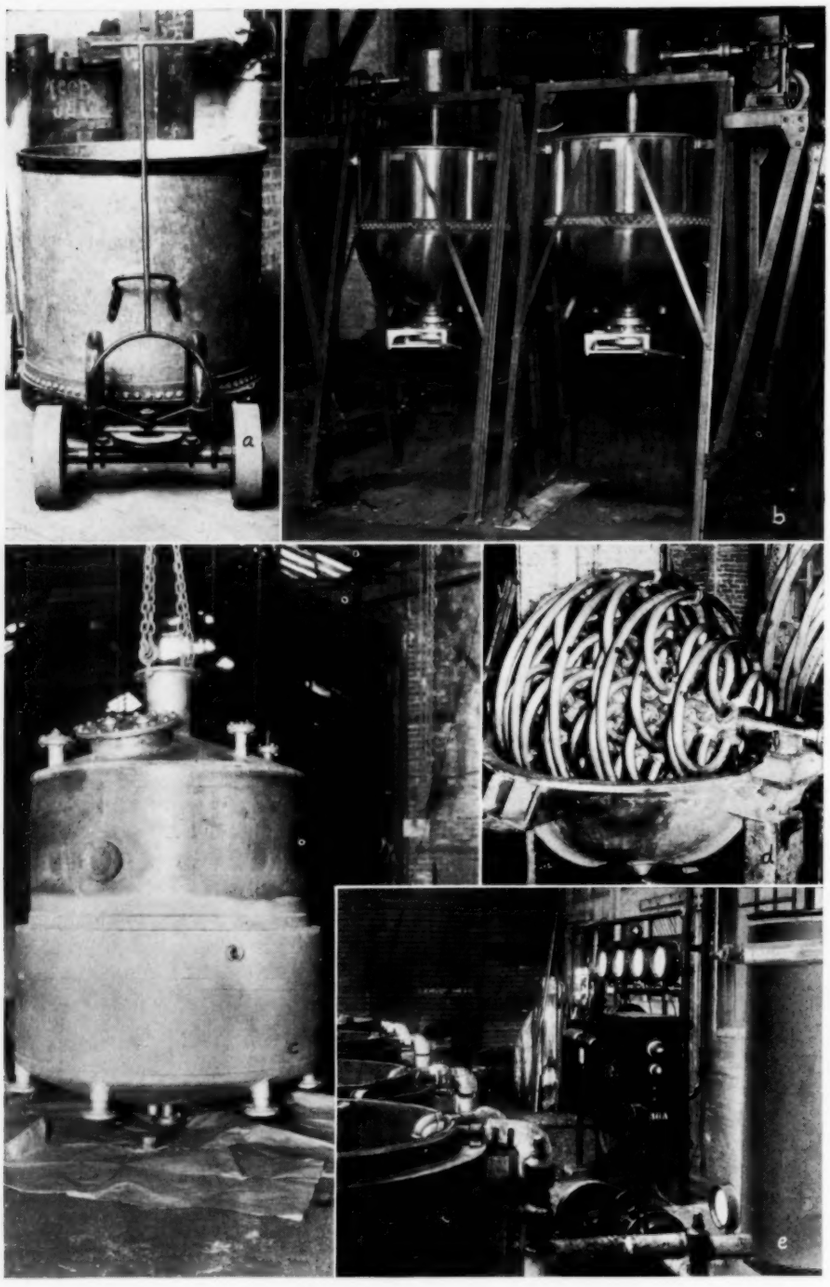
[1]Among the industries which use lard, tallow, fatty acids and similar materials are bakeries, soap works, packing plants and chemical works of many kinds. Resins, shellac and gums are used in the manufacture of paint, varnish, linoleum and naval stores. Pitch, sulphur, waxes and other compounds are used in the manufacture of textiles, paper, rope, roofing and road materials, and for many other products.



a One of the oldest and simplest methods for heating batches of heavy liquid by direct firing is an open vessel mounted on wheels, as shown here. This type of equipment has long been used for making varnish, boiling linseed oil, and in similar processes. The kettle is rolled over an opening in a furnace fired by coal, oil or gas. During the cooking process, the material must be carefully stirred by hand to avoid scorching. This direct firing involves considerable labor and skill and with some materials presents a hazardous fire risk in case they boil over.

b A pair of large kettles having mechanical stirring equipment and jacketed so they can be heated by steam or circulating hot oil. They are constructed of pure nickel to avoid corrosion. The large, free opening valves at the bottom permit dumping thick, heavy materials into containers placed underneath.

c A welded monel pressure or vacuum vessel for heating chemicals; the lower portion is jacketed so that it may be heated by steam or by circulating hot oil according to the temperature desired.

d Some materials are too poor thermal conductors to be heated by means of a jacket around the vessel, and stationary heating coils in the material itself are not satisfactory because the material will not circulate and the heat will not penetrate evenly throughout the mass. This view illustrates a unique design of a closed pressure vessel for heavy wood pulp compounds used in the manufacture of tanning reagents. The vessel shown has a cast iron shell lined with seamless sheet copper. Steam or hot oil is circulated through the assembly of copper coils which is connected to the hollow shaft. The shaft carrying the heated coils is rotated slowly and the coils are so shaped that every portion of the mass of material within the container is thor-

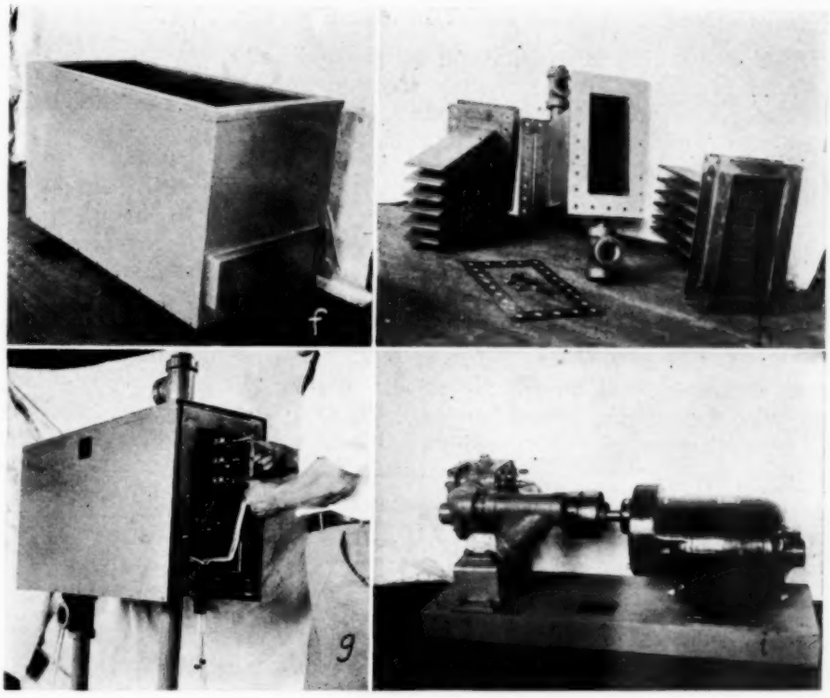
about 55 per cent of the total heat energy, which does not cause any increase of temperature.

Local Reheating of the Piping

Because "Class 1" materials remain fluid through a wide temperature range, it is possible for a suitable pump to maintain flow after it has been started, even though the material may cool somewhat in the pipe lines. With "Class 2" materials however, owing to their rather narrow range of critical temperature, there is always danger of complete re-solidification at any point in the pipe line which becomes sufficiently chilled. The flow may be entirely blocked and the whole line become filled up with solid material; therefore local re-heating at certain points frequently is necessary. Provision must be made for completely draining the pipe lines at the end of each operation or for pre-heating all the material remaining in the pipe lines before pumping is again started.

Tanks, containers, pipe lines, pumps, fittings, and so forth, can be heated by circulating steam or hot oil through jackets and special passageways provided for that purpose. Where steam is not readily available or for temperatures above those that can be obtained with steam at safe pressures, hot oil is preferable. This hot oil must be kept circulating by pumps and can usually be most successfully heated by electricity. In some cases of local re-heating, electricity can be applied directly to pipes or containers without the intermediate use of circulating oil as a heat transfer agent. In such cases, care must be used properly to insulate the heaters both electrically and thermally. They must also be safeguarded from being saturated from leaks which may occur. In

oughly stirred by their rotation. In other words, the heating coils themselves serve as stirring paddles.—*(Photo courtesy Jos. Oat & Sons, Philadelphia.)*



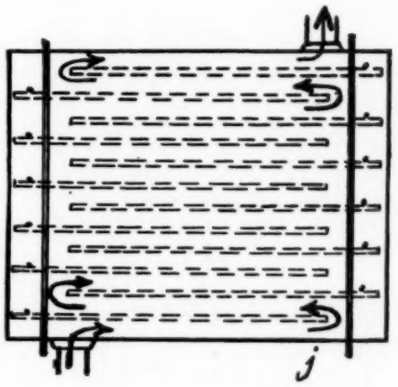
e An installation of large vats set in the floor. They are heated by a pumped circulation of hot oil, heated electrically. The electric oil heater is shown at the extreme right and the control panels and temperature recording instruments appear in the background.

f A rectangular type of insulated compound tank; the heat insulation around the tank is covered with a welded steel jacket to prevent its saturation by material dripping over the sides and to protect it from damage by rough usage.

g Close-up view of a circulating type of electric oil heater. The heating elements are not immersed in the oil but slide snugly into hollow, flat, sealed tubes; all the electric connections are therefore in a compartment sealed from the oil, yet readily accessible through a special door closure.

h A heater similar to that shown in "g," dis-assembled. A battery of tube sections is inserted from each end and made tight with a suitable gasket. The tubes slide into grooves on the side walls and are spaced alternately so that they form a continuous passageway as shown in "j."

i Typical pumping unit for circulating hot oil. The pump must be designed to operate at high temperatures and must be capable of pumping heavy, viscous oil when starting up. The working parts and stuffing boxes must be suitable for high temperatures. The pump should be driven directly by a gear reduction motor through a flexible coupling. It is very important that the electrical power used for heating the oil be interlocked so that it can not be turned on unless the pump is running and so it will automatically be turned off if the pump is stopped for any reason. It is also important that suitable, easily cleaned, large area strainers be used at the pump intake, to intercept any foreign matter that would damage the pump. In any closed system, an expansion tank and pressure relief safety valves are required.



j This passageway has a flat, rectangular cross-section, and its walls form large heating areas of low unit heat density. The uniform, positive flow of a thin, flat stream of oil over every part of the heating surface avoids hot spots and carbonizing. (See "h").

heating inflammable materials, it is essential to avoid all fire risk by taking proper precautions in design.

Cooling Needed at Certain Points

In addition to heating "Class 2" materials, it also is often necessary to provide cooling at certain points—such as around the stuffing boxes of pumps. Many materials (for instance, resin and waxes), have no lubricating qualities and set so solidly around the packing in the stuffing boxes that unless the material is melted at that point before the pump is started, the packing will be torn loose and destroyed. On the other hand, when flow is started and the hot material is passing through the pump, it may be so thin at the handling temperature that it is difficult to prevent leaks at the stuffing boxes This is aggravated by the lack of lubricating quality of the fluid. Trouble can be overcome by changing from heating the stuffing box to cooling it as soon as the flow of hot liquid begins. The temperature at the stuffing box should be kept such that the material will be just fluid enough to prevent seizing the packing and yet not thin enough to cause troublesome leaks. This close temperature control is necessary because of the narrow, critical range of temperature between solid material and an extremely thin fluid.

A large pump handling hot resin is shown in one of the cuts. Steam is circulated through passageways for heating-up purposes, and after operation starts, water is circulated to give the right amount of cooling to maintain the stuffing box temperature at the correct point. Where the heating-up is done by means of hot oil instead of steam, the circulation is changed to use cool oil, the latter passing through a heat exchanger which is water cooled. In other cases, two separate and independent passageways are provided, one for cooling by water.

Piping, Tanks, Must Resist Corrosion

Different materials attack different metals, and it is important to use metals which resist corrosion, especially at critical points such as pumps, valve seats, and so forth. Fatty acids and soaps tend to attack iron and steel. Bronze, monel, chrome steel, nickel, Everdur, copper or aluminum are desirable. Pumps should preferably be of bronze with bronze or chrome steel shafts. Materials such as resins, paints, varnishes, and many other products tend to attack copper and bronze but chrome steel, monel, nickel, aluminum, and in some cases iron or steel are satisfactory. Lard, tallow, paraffin, and many waxes, are neutral and do not attack metals. The design of tanks, containers, and mixers for storing, heating or processing these various materials requires careful study.

The development of welding and brazing these special metals has made it possible to extend their use into the range of higher temperatures and pressures, and refinements in heat application have further extended the field for certain materials. A good example of this is aluminum, which is a very desirable metal for containing certain fluids but because of its low tensile strength at elevated temperatures, can not be used for high pressure steam jacketed vessels. However, by utilizing hot oil as a heating medium, the temperature range in which aluminum vessels can be used is raised considerably, for the hot oil circulated through the jackets requires only a few pounds' pumping pressure because its temperature is not related to its pressure as is the case with steam.

The illustrations have been selected to show various methods by which materials are handled, and are fully explained in the captions.

Drawing of windowless department store being built by Sears, Roebuck and Company. First to fourth floors are for sales, fifth floor for offices and storage, main basement for food department

Windowless Construction Reduces Air Conditioning Load

....and has other advantages, according to Sears, Roebuck. New store shows influence of World's Fair on architecture.

ENTIRELY windowless except for four slim spires of glass rising 58 feet above each entrance, the new Sears, Roebuck and Company department store being erected in Chicago at 63rd and Halsted streets demonstrates the influence of the architecture introduced by A Century of Progress exposition last year. Five stories in height and with basement and sub-basement, the building's foundations and columns will allow for three extra floors when expansion becomes necessary. All sales areas are to be completely air conditioned the year around, the electrical load for air conditioning being estimated as 465 kilowatts.

The windowless construction is used because of the practical advantages it achieves. According to L. B. DeWitt, Sears construction head, these include the simplified application and economical operation of the air conditioning system, as it is expected the load on the air conditioning plant will be considerably reduced; savings in merchandise losses by eliminating seepage of dirt and smoke through windows; elimination of deterioration of certain goods by sun-light; banishment of street noise; and uniform lighting, with consequent opportunity for greater power and conviction in merchandise displays. Six thousand square feet of floor space along the walls of upper floors will be saved for displays and storage.

The columns of glass over the entrance also serve a strictly utilitarian purpose; they will admit light into the ends of the center aisles for the examination by the customer of style merchandise and piece goods before purchasing.

DECREASING TOTAL STEAM CONSUMPTION per year despite a steady increase in building cubage served is the record of the University of Chicago's department of buildings and grounds. Efficient management, equipped with steam flow meters (119 of them) to show where the steam is going, is a good part of the answer. Readings made twice a month are studied carefully, opportunities for saving steam are considered. Heat loss is reduced by weather-stripping, caulking window frames, etc.; steam is saved by revising operating schedules, controlling temperature automatically..... this article presents significant data, explains how economy is obtained, suggests what can be accomplished by alert management with the proper equipment and instruments

Alert Management, Meters, & Proper Control
CUT BUILDING HEATING COSTS

By **Lester S. Ries***

A SERVICE department of an educational institution does not have the usual means of gaging its effectiveness by means of watching the profit-and-loss account keep in the black as is the case in industry. Other methods of necessity are adopted to tell the story so that management can competently control its operation and determine its contribution to a balanced budget.

A most significant indicator is a comparison of the handling and consumption of steam in the plant for successive seasons, reduced to comparable units of consumption in order that a just comparison can be made. At the University of Chicago, the Department of Buildings & Grounds is charged with the generation and distribution of steam. Steam is produced in four 1200-hp boilers and conducted to the campus in two 12-in. mains located in a 7 ft x 8 ft concrete tunnel 5,200 ft long, at a pressure of 150 lb.[1] There are two miles of secondary distribution tunnels in which the steam pressure is 75 lb gage. This pressure is again reduced in the buildings to that desired for heating, cooking, sterilizing, etc.

119 Steam Meters Show Building Steam Consumption

This central steam plant was placed in operation in August, 1929, together with the 10,400 ft of high pressure distribution mains. Steam metering devices were installed at the point of entry into the high pressure tunnel mains, at the branches from the main tunnel to the secondary distribution tunnels and at the points of steam consumption. In all, there are 119 steam meters in use. By means of this complete metering installation, all the steam delivered by the boiler plant is known and the ultimate consumption at the buildings is measured.

Use of the Meter Records

In order to show the use made of the information collected by means of this equipment, it is desirable to include the following significant statistics:

Year	Bldg. Cubage Served M cu ft	M-pounds Steam to Tunnels	Degree Days for Heating Season	Pounds of Stm. per D.D. per M cu ft
1928-29	34,104	535,146	6,122	2.56
1929-30	36,607	513,515	6,318	2.22
1930-31	39,232	456,824	6,048	1.93
1931-32	45,519	401,462	5,244	1.68
1932-33	51,409	402,806	6,087	1.29
1933-34	51,409	389,441	6,091	1.24

This tabulation includes all the steam delivered to the distribution system by the heating plant.

It will be noted that there is a consistent lowering of the figure shown under the heading "M-pounds per Degree Day per M cu ft of Building." This figure is the unit used for comparison month by month and year by year.

Records are kept of each building and the unit of consumption for each month is compared to the figure of the corresponding month of the previous years. By this means the standard or normal for each individual building is established and the peculiarities and characteristics of the buildings can be watched.

How Steam Is Saved

Meters are read twice a month. With this information available, an engineering study is made of buildings in which the consumption of steam was high and in which it is thought economies can be made. As a result, in

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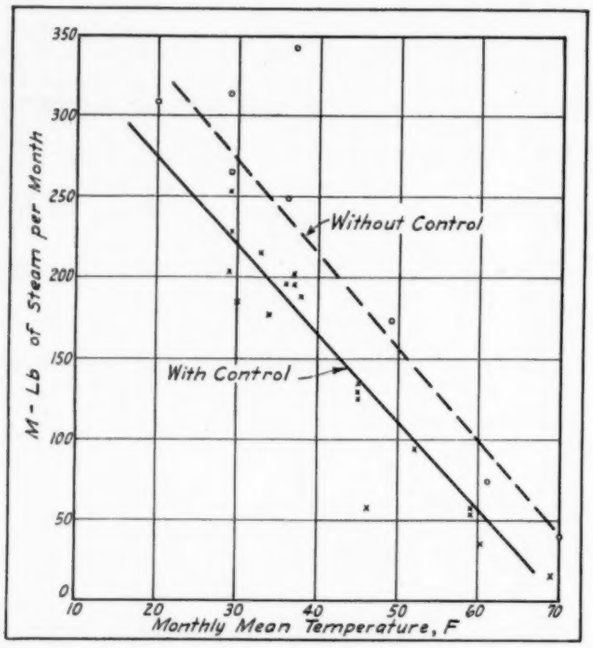
[1]See H. P. & A. C., p. 90, February, 1933.



Steam consumption is reduced by the installation of automatic zone control. This chart refers to the Walker Museum, a typical University building. The control installation is described in the text, as are the other methods by which the steam requirements of the building were reduced

many cases one or more modifications to the building have been made in order to decrease steam consumption. These have included such things as:

(a) Revision of operating schedule for heating the building.
(b) Changing single pipe installations to two-pipe vacuum return.
(c) Changing to more efficient type of radiator air vents in single pipe systems.
(d) Simplifying building piping.
(e) Improving steam pipe insulation.
(f) Adding temperature control and steam flow control devices.
(g) Weather-stripping.

Walker Museum a Typical Case

To illustrate, let us consider one of the buildings—the Walker Museum. This building, constructed in 1893, is of mill construction with Bedford limestone masonry walls typical of the University buildings. It is exposed on the north, east and south elevations and joins Rosenwald Hall on the west. It is used as a geological museum and workshop and has three floors and a basement, all in use. Significant data are:

(a) Cubage, 442,356.
(b) Floor area 25,484 sq ft.
(c) Single pipe radiation, 3,100 sq ft.
(d) Windows, 146 (including window transoms.)
(e) Ceiling heights—Basement 9'2"; first floor, 18'5"; second floor 16'5"; third floor, 15'3".

This building, it will be noted, has a large number of windows, and in 1929 **metal weather-strips were installed,** largely eliminating heat leakage from this source. **Caulking window frames** is considered a routine matter of maintenance and is given attention as required. Wherever possible, in caulking, the window frame brick moulding is removed so as to apply oakum and caulking compound most effectively, and then re-installed. This results in reducing the frequency for this need and makes for a much neater building exterior than if the caulking is done in the common way, with a "gun," or trowelling the compound in from the exposed opening between the window frame and the stone.

Temperature Control Saves 38%

The building is supplied with a 2-in. high pressure main furnishing steam through the south building wall at 75 lb gage. The steam for heating purposes is reduced in one stage to the building pressure. From the reducing valve the steam supply divides into two mains, one 4-in. and one 5 -in. Each of these low pressure supply mains serves a section of the building, the 4-in. serving the south elevation and the 5-in. serving the north and east exposures. All the steam piping and returns are insulated with 85 per cent magnesia covering in the basement, but the risers throughout the building are bare.

Two motor-operated thermostatically controlled steam valves were installed in the steam supply, one in the main supplying the radiation on the south elevation, and one in the main supplying the north and east exposures. Clock thermostats were installed in strategic locations, one on the first floor operating the valve controlling the steam supply to the southern exposure and one on the second floor for the other valve. The building was operated so that there were alternate periods when the devices were "on" and "off," over a period of about three months. These periods were of varying length from one day to a week; the period uncontrolled equaling the period of control following. The results were as follows:

	CONTROL "OFF"				CONTROL "ON"		
1931	No. Days	Lb of Steam	Lb / D.D. per M cu ft	1931	No. Days	Lb of Steam	Lb / D.D. per M cu ft
Jan.	13	155,900	0.935	Jan.	18	139,500	0.495
Feb.	14	151,200	0.779	Feb.	14	92,100	0.535
Mar.	14	139,700	0.776	Mar.	17	118,500	0.506
Total	41	446,800	....	Total	49	350,100	....
Mean	..	.....	0.830	Mean	..	.....	0.512

This shows a reduction of 0.318 lb of steam used per degree day for each 1000 cu ft of building, with the thermostatic operation of the steam supply valve controlling the supply than when the building was heated with the manual operation of the steam supply valves. This reduction is about 38 per cent. The residual heat in the building walls and contents affects this tabulation to some extent, tending to reduce this differential. The temperature in the building was kept at at least 70 F during hours of occupancy, from 8 a. m. until 10 p. m.

Heating *and* Air Conditioning Buildings *with the* Reversed Refrigeration Cycle

A GAS-ENGINE-DRIVEN compressor or heat pump for heating buildings by the reversed refrigeration cycle—the same equipment being used for cooling in summer—is the proposal of the author, who shows in this article the arrangements that might be utilized, discusses possible objections to the scheme, brings up the questions of cost and practicality..... In his article in the June issue the principle of heating by the reversed refrigeration cycle was explained, and one type of system described; this article concludes the discussion.

By **C. F. Mowrey***

THE use of a gas-engine-driven compressor for heating buildings by the reversed refrigeration cycle (or heat pump) was proposed by the author in an article in the June issue of H.P.&A.C. The arrangement includes the paralleling of condensers and evaporators, the air being warmed in steps as it is forced through one condenser after another. By proper adjustments the same equipment may function as a cooling plant during the summer.

In addition to a discussion of the principles involved the first article described the application of this method to a warm air type of system, all air being recirculated. This month the application to other types of systems, costs, possible objections, etc., are considered.

Warm Air Type of System, No Recirculation

There are many types of buildings, particularly public buildings, where the ventilation requirements are such that if the amount of outdoor air required were heated to 120 F or less, enough heat would be supplied to maintain the proper temperature within the building. We will analyze a typical case to point out the efficiency that might be expected.

Assume that the heat loss from a building is 100,000 Btu when the indoor temperature is 70 F and the outdoor temperature is zero and that 125,000 Btu per hr would be supplied by the heating plant if the return air entered the plant at 65 F. In order to strike an average efficiency we will assume that the outdoor temperature is 30 F. The heating plant would be required to supply heat enough to heat the air from 30 to 125 F, which quantity would compare with the 125,000 Btu as the difference between 125 and 30 compares with the difference between 125 and 65. Therefore, the heating plant would be required to deliver 125,000 Btu $\times \frac{125-30}{125-65}$ or, 198,000 Btu.

On a cut-and-try basis, we will assume that 65 per cent of the heat, equalling 128,800 Btu, would be supplied by the heat pump. Then the incoming air temperature would be raised 65 per cent of the way from 30 to 125 F or 63 F, to a temperature of 93 F. It would be in steps of from 30 to 51, 51 to 72 and 72 to 93, with condenser temperatures at 61, 82 and 103 F, the average being 82 F. The coefficient of performance would be determined by the average condenser temperature of 82 F and the evaporator temperature of 20 F (10° below 30°). This coefficient would, therefore, be equal to $\frac{460+20}{82-20}$ or 7.75. As previously explained, the actual coefficient of performance would be considered as only 15 per cent of the theoretical, or 1.16. As the 128,800 Btu would equal 1.16 times the amount of heat supplied the engine, the amount supplied the engine would be $\frac{128,800}{1.16}$ or 111,000 Btu.

If we estimate, as before, that 60 per cent of the waste heat could be utilized, we could add 60 per cent of 111,000, or 66,600 Btu to the 128,800 Btu delivered by the heat pump, making a total of 195,400 Btu delivered. We started with the requirement that 198,000 Btu be delivered, which shows that we did not pick up the right percentage of the heat that would be supplied by the heat pump. Actually, a little less than 65 per cent of the heat would be supplied by the heat pump.

The resulting "efficiency" for this case would be equal to $\frac{195,400}{111,000}$ or 176 per cent, which means that for these conditions this type of plant would require only 43 per cent as much fuel as would a gas furnace of 75 per cent efficiency. If we had picked the exact percentage of heat to be supplied by the heat pump, the efficiency would have been practically the same. In some types of buildings the air could be supplied at a lower temperature, which would increase the efficiency further.

As a matter of fact, it would be more desirable to have this type of plant operate at least two-thirds of the time, at a capacity corresponding to the heating requirements for such operation. The above conditions of operation were chosen so that a direct comparison could be made between the first case, considered in the previous article, based on recirculation of air with an indoor temperature of 70 and an outdoor temperature of 30 F (discussed in the June article), and this second case, based on the same

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temperatures but no recirculation of air. It may be noted that, whereas, if a conventional furnace were to be used, 58 per cent more heat would be required to be delivered in the second case than in the first case, the above figures show a required increase of only 35 per cent for the heat pump method of heating. The input for the recirculation case was 82,500 Btu per hr, whereas the input for the fresh air case is 111,000 Btu per hr.

Let us now assume another "fresh air" case for 70 F indoors and zero outdoors. We will again assume that for such conditions and recirculation, the hourly heat loss would be 100,000 Btu and that it is required to supply 125,000 Btu per hr. However, if no air were to be recirculated, the amount of heat required per hour would bear the relation to 125,000 that the difference between 125 and zero bears to the difference between 125 and 65. On this basis the amount of heat required would be 260,000 Btu

$$\left[\frac{125-0}{125-65}\times 125{,}000\right]$$

We will assume on a cut-and-try basis that 67 per cent of the heat, equalling 174,000 Btu, would be supplied by the heat pump. Then the pump would be required to raise the temperature of the outdoor air 67 per cent of the way from 0 to 125 or to 82 F. This could be done in three steps, *i.e.*, from 0 to 27, from 27 to 54, and from 54 to 82, by the three condensers, each at a temperature 10 F higher than that of the air (see diagram). The average condenser temperature would be 64 F. Now the efficiency of the plant can be increased by arranging to have the exhaust air blow through two of the evaporators on the way out. If the air were exhausted at 0 F it would have given up 135,000 Btu

$$\left[\frac{65}{125}\times 260{,}000\right]$$

of the 174,000 required. If the evaporator temperatures are 33, 9 and —10 F, the proper amount of heat will be picked up at an average of about 10 F. The C. P. would then be

$$\left[\frac{460+10}{64-10}\right]$$

or 8.7. The C. P. based on the heat supplied the engine would be 8.7×0.15 or 1.30.

As the heat supplied by the heat pump would be 1.30 times the heat supplied the engine, the heat supplied the engine would be $\frac{174{,}000}{1.30}$ or 134,000 Btu. The heat available as waste heat would be 60 per cent of 134,000 or 80,300. Therefore, the total amount of heat delivered would be the sum of 174,000 and 80,300 or 254,300.

The stated condition was for a delivery of 260,000 Btu per hr, which shows that the assumption that 67 per cent of the heat would be supplied by the heat pump was not quite correct. For the above conditions, the temperature of the heating air leaving the plant would be about 122, instead of 125 F. A temperature of 122 for this type of heating would probably be high enough. As a matter of fact, air at lower temperatures is customarily supplied certain types of buildings.

The "efficiency" for this case is equal to $\frac{254{,}300}{134{,}000}$ or 190 per cent. It may be noted that the heating value of the fuel required at the engine was 134,000 Btu per hr contrasted with 100,000 Btu per hr for the recirculation case. This is an increase of only 34 per cent. It is probable that the actual increase would be much less for the following reason: When a building is rated at a certain Btu loss per hr, it includes the heat loss chargeable to at least one air change per hour. This loss usually amounts to somewhere in the neighborhood of 15 to 20 per cent. In buildings where none of the air is recirculated, it is not necessary to open doors or windows for fresh air, and in addition precautions are usually taken to prevent infiltration. These factors should reduce considerably the 34 per cent difference.

Although the "fresh air" plant might be operated nearly as cheaply as the recirculation plant discussed in the first article, due to the cost of the equipment, it probably would not be advisable to use the former unless required for proper ventilation. According to our figures the recirculation case would require a 6.33 ton capacity (standard rating) heat pump, a 10 hp gas engine and coils sufficient to absorb and dissipate 125,000 Btu per hr, whereas, for the fresh air case there would be required a 13.4 hp engine, a correspondingly larger heat pump and coils enough to absorb and dissipate 260,000 Btu per hr. One advantage in favor of the fresh air case, however, would be that the exhaust air from the building might occasionally be diverted through the low temperature evaporator to melt accumulated frost.

Other Methods of Application

In any location where well water is available, these efficiencies could be exceeded considerably. The temperature of well water is always close to the average yearly air temperature, which in the center of New York State is about 48 F. Also, the transfer of heat between water and metal is effected much easier than between air and metal. The power required to pump the water would be slight in comparison with the increased efficiency. This latter fact is proved constantly in the operation of refrigeration plants.

The heat pump method of heating could also be used in connection with a hot water system of sufficient radiator capacity. The water would pass through the three or more water-cooled condensers, each of which would raise the temperature of the water, then through the jacket of the gas engine and finally through the heat exchanger it would absorb heat from the exhaust gas.

Still another application which might prove its usefulness would be a comparatively small plant to be used for spring and fall heating of a residence and winter heating of a garage. This plant could be located in the garage and connected to both the water system in the house and a radiator in the garage. It could be used to heat the house in the fall until the weather grew quite cold and could then be used to heat the garage. In the spring it could again be used to heat the house. In the summer a manipulation of valves would interchange the evaporator and condenser and it could then be used to supply cold water for cooling all or part of the house. Due to the small temperature differences between the outside temperature and the required temperature in the house in spring and fall, and the required garage temperature in winter, the efficiency of such a plant would be very high. Of course, it would be necessary to incorporate a fan so that the engine would be air-cooled in the summer.

Possible Objections to the Method

Let us consider some of the arguments that might be used against the advisability of the development of the gas driven heat pump method of heating. First is the

question of cost. As was previously stated, it is expected that summer cooling and air conditioning are going to continue to be rapidly accepted. What must be determined, then, is not the cost of a complete gas-driven heating and cooling plant, but the difference between such a plant and the total cost of a separate cooling plant and a conventional type of heating plant.

A complete electrically driven cooling system would include a compressor, a motor, one or more condensers and one or more evaporators. A gas driven combination plant, as described above, would include a gas engine and compressors on one base, three condensers and three evaporators. If three condensers and evaporators were to take the place of but one each, and the gas driven compressor were to take the place of a compressor only, the gas driven equipment would, of course, cost more. Considering, however, that it would not require a large motor, if any, for starting and that the present cost of the two cycle fuel injection gas engine is quite low, indications are that the total difference in cost between the electrically driven cooling system plus a heating plant and the combination gas system, including necessary controls, would be low enough to make it a good investment.

If the plant were to be installed in a new building, it would not be necessary to construct a chimney to carry away the exhaust gas, as an insulated pipe would serve.

The next question is the length of life of a gas engine and compressor. A check-up of gas engines and compressors will show that a great number have exceeded by far, in hourly operation, the equivalent of the number of hours that a combination plant would operate in twenty years.

Then, there is the question of noise. For many of us, the mental picture of a stationary engine is associated with plenty of racket. These engines are noisy chiefly because the ordinary engine is used where a certain amount of noise is permissible. The manufacturers say a gas engine can be satisfactorily muffled. It might present quite a problem, but it can be done.

It would not be very difficult to try this system out for a season, by using equipment now available. There are a great number of commercial buildings using high priced fuel, to which the warm air is supplied at a comparatively low temperature. Incidentally, many of these buildings require several air changes per hour, and are ideal for the application of such a system. Large-sized gas-driven compressors are now available, as well as all shapes and sizes of evaporators and condensers. Consequently, a complete plant could easily be put in operation. It could be operated manually by some person competent to make a complete report of its operation, from which valuable information could be obtained to aid in the development of completely automatic plants.

Although this article has made continual reference to a gas-driven compression heating system with an assumed coefficient of performance of 15 per cent of the ideal, it is to be borne in mind that any absorption or adsorption system that could be made to produce an equal coefficient of performance could undoubtedly be used to as good an advantage.

The rapid acceptance of air conditioning despite the depression is ample evidence that there is a good field for some of the systems described above, in the author's opinion. The public is now familiar with engines and will certainly accept mechanical equipment which has the combined appeal of improved living conditions with economy.

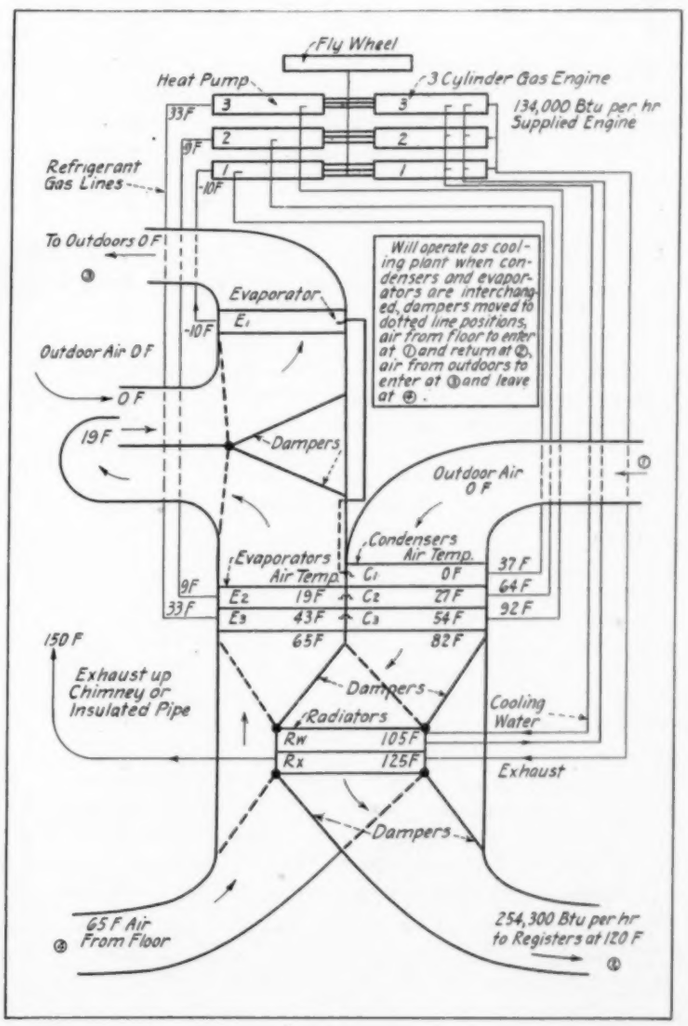

Diagram illustrating one of the examples worked out in the text. "Efficiency" is 190 per cent.

Requirements for

Welded Piping Joints

Welding boiler feed piping

.....as given in the proposed American Standards Association ***Code for Pressure Piping are explained and discussed in this article; the next article in this series will include quotations from the Code's section on welding. The author gives this part of the Code detailed treatment because of the importance of welded joints in present day piping systems.***

By **Sabin Crocker***

THE importance of welded joints in present day piping systems is deemed to justify thorough treatment of that topic in this series on the Proposed Code for Pressure Piping. Chapter 3 of Code Section 5 on fabrication details covers the welding of pipe joints. In that chapter terms relating to welding are defined, approved details of design and procedure are described, acceptance tests for the completed product are set forth, and qualification tests for welders are established.

Fusion welding is defined by the code as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Both gas and electric-arc welding come within its scope. No direct stipulations are made regarding the use of the carbon arc, bare or coated metallic electrodes, protecting atmospheres or the like since the code requirements for quality of finished work in themselves are deemed to constitute sufficient protection.

In preparing its chapter on welding the sectional committee on the code for pressure piping carefully considered the requirements for welding already set up by various other committees and organizations. Among the codes and rules for fusion welding so considered were those of the *A.S.M.E.* boiler code committee for boilers and unfired pressure vessels; the joint *A.P.I.-A.S.M.E.* committee on the design and construction of unfired pressure vessels for petroleum products; the *American Welding Society* for welding structural steel; the Hartford Steam Boiler Inspection and Insurance Company for pressure piping; and the *American Gas Association* for pipe joints. These organizations either co-operated as a group with Sectional Committee *B31* in formulating its requirements for the welding of piping joints, or had representatives on *B31* who directly participated in drafting its rules. Notwithstanding the care and thought exercised by the sectional committee in formulating its rules for fusion welding, some points still remain in controversy and will have to be ironed out through a process of seasoning and compromise. This situation is to be expected in attempting to standardize procedure for an art which is developing so rapidly and in so broad a field as is fusion welding.

Basic Methods for Securing Sound Welds

The two basic methods employed in all pressure-vessel or pipe welding codes as a means for securing sound welds are: (a) the periodic qualification test for welders and (b) the non-destructive examination of each completed weld. Either one or both methods are called for as circumstances warrant. Occasionally other methods of ascertaining soundness are resorted to for purposes of check, among which are the trepanning or cutting out of a coupon to permit examination of a doubtful weld, and the unexpected removal from the line and cutting up of an entire weld for strength and ductility tests, observation of penetration, freedom from porosity, etc. For the purpose of this discussion the adjective "sound" is used as embracing all the good qualities which should be found in a weld. According to Webster's dictionary "sound" means: "(1) Free from flaw, defect, or decay; not impaired.—— (3) Firm; Strong; safe; ——." When a weld meets predetermined standards for all these properties, what more could be desired?

Qualification Tests of Welders

In the absence of adequate and inexpensive non-destructive tests for completed welds, there is a general conviction that the operator should be required to demonstrate his qualifications through rather elaborate tests before being allowed to strength-weld joints in pressure piping. Seal welds are not considered as contributing to the strength of joints and no welding tests are required

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Table 1—Summary of Code Requirements for Welded Joints of Piping and Pressure Vessels

Code / Service	Maximum Service Conditions: Pressure Psi	Maximum Service Conditions: Temp. F	Longitudinal Seams	Transverse Seams and Heads	Branch Opening Welds	Test Plate	Qualification Test for Welders	Stress Relief	Non-Destructive Tests: X-Ray or Gamma-Ray	Non-Destructive Tests: Hydrostatic and Hammer	Non-Destructive Tests: Straight Hydrostatic	Trepanned Coupon	Tensile Strength, Specimen psi Min.	Elongation of Tensile Spec., Min. % in 2 In.	Elongation on Free Bend Test, Min. %	Nick Break Test	Joint Effic., %	Specific Gravity Min.
A.S.M.E. Boiler Code—Power Boilers Class 1	No Limit		Double-Welded Butt	Double-Welded Butt	Butt or Fillet	Yes	None	1100 to 1200 F	Either	1½ WP[22]	2 WP	No	Same as Plate	20	30	No	90	7.80
A.S.M.E. Code for Unfired Pressure Vessels Class 1	No Limit		Double-Welded Butt	Double-Welded Butt	Butt or Fillet	Yes	None	1100 to 1200 F	Either	1½ WP	2 WP	No	Same as Plate	20	30	No	90	7.80
Class 2	400[1]	700[1]	Double-Welded Butt	Double or Single Butt[2]	Butt or Fillet	No	6 mo.[3]	1100 to[4] 1200 F	No	1½ WP	2 WP	No	95% of Plate	No	20 Elec. 15 Gas	Yes or X-Ray	80	No
Class 3	200[5]	250[5]	Note[6]	Butt or Fillet	Butt or Fillet	No	6 mo.[3]	Certain Nozzles Only	No	1½ WP	2 WP	No	42,000[7]	No	10	No	Note[8]	No
Proposed A.P.I.-A.S.M.E. Rules for Unfired Pressure Vessels for Petroleum Products Highest Grade Fusion Welding	Depends on Diam., 1000 etc.		Double-Welded Butt Note [19]	Single or Double Butt	Full Depth of Plate	1 for each 100 ft. weld	6 mo.[9]	Over 1100 F	Either	1¼ WP	1½ WP	No	Same as Plate	20	30	Yes or X-Ray	95	No
Second Grade Fusion Welding	Depends on Diam., 750 etc.			Single or Double Butt	Full Depth of Plate	1 for each 100 ft. weld	6 mo.[9]	No[11]	No	1¼ WP	1½ WP	No	Note[12]	No	20	Yes or X-Ray	Note[13]	No
Hartford Steam Boiler Insp., etc. Piping Joints	No Limit			Butt or Fillet	Butt or Fillet	No	Yes		No			No	90% of pipe	No	20 Elec. 15 Gas	Yes		No
Piping Joints	200	250		Butt or Fillet	Butt or Fillet	No	Yes		No			No	85% of pipe or 42,000	No	10%	Yes		No
Proposed A.S.A. Code for Pressure Piping Highest Grade Fusion Welding	No Limit		Note[14]	Butt or Fillet	Butt or[15] Fillet	No	6 mo.	1100 to 1200 F[16],	No	Note[17]	Note[17]	Yes	90% of pipe	No	15% Note[18]	Yes	90 Longitudinal Seam[14]	No
Second Grade Fusion Welding	See Table 2[21]		Note[19]	Butt or Fillet	Butt or[15] Fillet	No	6 mo.	1100 to 1200 F[16]	No	Note[17]	Note[17]	Yes	No	No	15% Note[20]	No	80 Longitudinal Seam[19]	No

Notes for Table 1

1. "All vessels covered by this Code may be included in this class, excepting those containing lethal gases or lethal liquids and/or those containing liquids operating at a temperature of 300 deg fahr, or above. The maximum pressure at which any vessel in this class may be operated is 400 lb per sq in., and/or the maximum temperature is 700 deg fahr, and the plate thickness as required by the permissible stress allowance shall not exceed 1½ in. This pressure limitation does not apply to vessels operated under hydraulic pressure at atmospheric temperature."

2. Circumferential joints on Class 2 vessels shall be of the double-welded butt type except for thicknesses of ⅜ in. or less, in which case they may be of the single-welded butt type.

3. Exception to this is allowable when the welder is regularly employed on production work embracing the same process and type of welding, in which case the tests may be effective for a period of one year.

4. "Class 2 fusion-welded vessels shall be stress relieved where both the wall thickness is greater than 0.58 in. and the shell diameter less than 20 in., and for other wall thicknesses and shell diameters, where the ratio of the diameter to the cube of the shell thickness is less than 100."

5. "All vessels covered by this Code, not exceeding ⅜ in. plate thickness and used for the storage of gases or liquids at temperatures not materially exceeding their boiling temperature at atmospheric pressure, and at pressures not to exceed 200 lb per sq in., and/or not to exceed a temperature of 250 deg fahr, may be included in this class, excepting those containing lethal gases or liquids."

6. "The longitudinal joints of Class 3 vessels may be of the butt-weld type for thickness of ⅜ in. or less, or of the double-welded lap type for thicknesses of ⅜ in. or less, or of the single-welded butt type for thicknesses of ¼ in. or less. If of the lap type the throat dimension of each of the welds shall not be less than $\frac{5}{4}T$, where T represents the thickness of the plate. Both edges of the lap shall be welded and the surface overlap shall not be less than $4T$."

7. "Tension Tests. Each full-section tension specimen should fail in the plate if the weld reinforcement is retained, but if failure occurs in the weld metal or along the line of fusion between the weld metal and the plate, then the tensile strength shall not be less than 90 per cent of the minimum of the specified tensile range of the plate used. For the reduced-section tension-test specimen the tensile strength shall not be less than 85 per cent of the minimum of the specified tensile range of the plate used. In no case shall the tensile strength be less than 42,000 lb per sq in."

8. Combined with the safe working stress of the material to obtain the "maximum unit joint working stress," which depends on the type of weld, viz double-welded butt, single-welded butt, double or single fillet etc. See Code for values.

9. A welder may be required to demonstrate his ability by test at any time during progress of the work.

10. Single or double-welded butt joint, or double-fillet lap welds. Plate ½ in. or more thick shall be double-welded.

11. Plate thickness shall not exceed 1¼ in. at the edge.

12. Ninety-five per cent of plate strength for all welds except single butt for which 80 per cent is allowed.

13. Fifty to 80 per cent depending on type of joint and thickness of plate.

14. Equivalent of Class 1 *A. S. M. E.* with X-ray examination and stress relief. Refers to *A. S. T. M.* specification being drafted for Class 1 Electric Fusion Welded Pipe. No qualification tests are required.

15. Nozzles, tees, and branches in materials ¾ in. or more in wall thickness shall be welded under shop or equivalent conditions.

16. Welded joints in carbon steel piping shall be stress relieved when the pipe wall thickness is ¾ in. or more and the service temperature is greater than 250 F.

17. Welded piping shall be given a hydrostatic test after erection where practicable. During hydrostatic pressure tests the pipe shall be hammered with sharp vibratory blows on alternate sides of the weld and close to it. The blows shall be hard enough to jar the weld without appreciably denting the pipe. Piping for 25 lb steam service pressure may be tested with air or water at 50 lb gage. A preliminary test with air at 100 lb gage may be made using soapy water to detect leaks, except for piping where the test pressure is less than 100 lb. Hydrostatic test pressures for other conditions are specified in the various Code sections at twice the service pressure, with the reservation that the test pressure shall not exceed 1½ times the rated service pressure of the valves and fittings and that the hydrostatic pressure shall in no case be less than 50 lb per sq in.; exceptions are: omission in the Oil Section of the 1½ times and the 50 lb minimum limits, and inclusion in the District Heating Section of a requirement that underground piping be subjected to a hydrostatic test of 1½ times the service pressure, to be maintained for at least 2 hours without sign of leak. Where this hydrostatic test is impracticable, underground piping may be tested with steam at service pressure before backfilling.

18. One bend-test bar of each pair shall show 15 per cent elongation without cracking when bent with the outside of the weld on the convex surface. The second bar shall bend 90° without breaking apart with the inside of the weld on the convex surface.

19. Refers to ordinary electric-fusion-welded pipe made to *A. S. T. M.* Specifications *A134 or A139*. No qualification tests are required.

20. Same as *Note 17* except two bars are required for each test.

21. To be published next month.

22. "WP"—working pressure.

to qualify a welder to seal-weld. Where test plates are provided for and complete X-ray or gamma-ray examination is contemplated, qualification tests are sometimes dispensed with as in the case of Class 1 boiler code welding. Due to its expense, the difficulty of field application, and the lack of significance of X-ray or gamma-ray examination of branch-connection welds, this type of non-destructive test has only a limited usefulness in connection with welded pipe joints. The use of test plates, which are welded as an integral part of Class 1 construction in making the seams of pressure vessels, boiler drums and the longitudinal seams of fusion-welded pipe, is not particularly applicable to welded pipe joints. Hence in pipe joint welding great importance is attached to periodic qualification tests, as is demonstrated by the summary of requirements from various codes shown in Table 1. The general belief is that qualification tests should be effective for a period of not to exceed six months, and be subject to further restrictions in that: (1) tests conducted by one employer shall not qualify a welder to work for another employer; (2) tests made with one welding rod or pipe material shall not qualify a welder to work with a different rod or material.

In the qualification tests of the code for pressure piping, a fundamental distinction is drawn between rolled welds and position welds. In rolled welds the work is done in the flat position, rotating the joint as the weld progresses so that the deposited metal is always applied on the upper side of the joint. In position welds the pipe is immovable and the operator is required to work in all positions around the joint. As described in the code some of the qualification tests are carried out on flat plates, welding in the downward or in the overhead positions as the case may be, while others are carried out on sections of pipe in either the fixed or free conditions with the pipe axis horizontal. Only those operators who have qualified for position welding are allowed to weld on that class of work. A schedule of the tests required for various classes of work is given in a table to be published with the next article of this series. The code contains a further requirement that nozzles, tees, and branches in materials ¾ in. or more in wall thickness shall be welded under shop or equivalent conditions, which means: rolling the work so that all welding is in the downward position; accurate cutting, fitting, and locating of each part; careful supervision of all welding; stress relieving of the entire nozzle, tee, or branch; adequate hydrostatic test of the completed fitting.

Check Tests of Welds

Where there is reasonable doubt whether a welder is maintaining the quality of work shown on his qualification test, the code for pressure piping provides that individual joints may be checked by trepanning out a coupon for a deep etch test, or in the case of small pipes by removing a section of pipe containing a weld to permit grinding and etching. These specimens are required to meet the standards for soundness and penetration specified elsewhere in connection with the nick-break test. Failure to meet this standard shall be cause for rejection of the weld tested. A second failure of a weld made by the same operator shall be cause for a check of all his welds, and for his suspension until he can show proof that his work fully meets the requirements for uniformly satisfactory welds.

Stress Relieving Welds

According to the code, welded joints in carbon-steel piping must be stress relieved, when the pipe wall thickness is ¾ in. or more, and the service temperature exceeds 250 F, by heating uniformly to temperature of 1100 to 1200 F which is held for a period of one hour per inch of thickness. Slow heating and cooling to and from that temperature is stipulated. This temperature corresponds to a dull-red heat and chiefly serves to reduce through creep any locked-in local stresses set up during the welding operation, the temperature and time specified having no appreciable effect on the grain structure of the metal. The necessity for and benefits derived from stress relieving are none too well established, varying to a considerable degree between methods of welding and the rate of application of weld metal as well as with the wall thickness. The principal changes to be expected in the physical properties of weld metal by stress relieving are improvement in the ductility and impact strength, both of which tend to increase the resistance of the joint to shock. In the absence of definite proof of the need for doing so, welded joints in high pressure and/or high temperature piping frequently are stress relieved even in thin walled pipe as an incidental precaution. It was not deemed advisable, however, to burden the code with requirements the need for which is not well demonstrated.

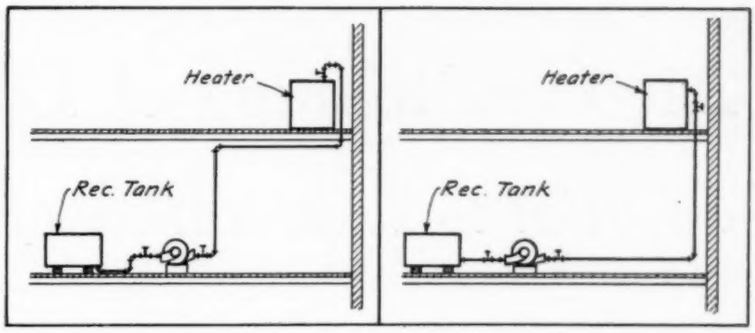
Alignment for Welding

According to code requirements piping and equipment shall be carefully lined up for welding so that no part is offset with respect to the adjacent part by more than 20 per cent of the pipe wall thickness. The problem of alignment involves something more than the mere lining up of parts, since the degree of ovality as well as the variation in wall thickness of the pipe has a bearing on the amount of offset at any point around a weld. Usual commercial tolerances in pipe manufacture permit plus or minus one per cent on out-of-roundness measured on the outside diameter of the pipe, and up to plus or minus 12½ per cent on wall thickness. Since the ovality tolerance mentioned above amounts alone to from 10 to 25 per cent plus or minus of the pipe wall thickness, misalignment due to irregularities in the pipe could easily exceed the specified maximum offset of 20 per cent of pipe wall thickness, *were such extreme variations common or were they cumulative at one point in a weld.*

The latest draft of the chapter on welding of piping joints is to be quoted in practically its entirety in the next installment of this series. While the requirements of the chapter are quite complete in themselves, they must be considered in conjunction with the requirements for materials and permissible construction laid down in other sections of the code for specific services such as power piping, gas and air piping, oil piping, etc. For instance, the welding chapter does not deal with the design of special manifolds fabricated from plate material, or with the manufacture and use of ready-made welding fittings, which are available both in seamless and in welded-plate construction. As mentioned before, the rules for fusion welding have been the subject of extensive discussion and undoubtedly will see further change before the code is adopted as an American Standard. After adoption they will still remain open to modification and amendment as is the case with other *A.S.A.* Standards and the *A.S.M.E.* Boiler Code.

BEFORE (left) the piping was changed the 8 elbows, 3 globe valves, the piping and elevation of the heater caused a total head on the pump of 43 ft of water, cut its capacity to 145 gpm

AFTER (right) the layout was revised, the total head on the pump was cut to 26 ft by using 2 elbows and 3 gate valves. Pump capacity now 240 gpm



By **N. T. Pef***

Pump Capacity Increased *by piping change*

MANY a piping job is done without rhyme or reason. Ordinarily, if the piping delivers the water or other fluid, does not leak or show any other visible imperfection, it is considered a success. The fact that it may take 10 per cent or more additional power for the pump over that required if the piping were arranged for minimum friction head is often ignored. Leaks particularly are not tolerated, yet the loss due to excess power required may be several times the loss due to leaks.

The diagram shows a "before and after" sketch of a job which illustrates my point. There were eight elbows on the original layout and only two on the other. The three valves shown on the "before" sketch are all globe while those on the remodeled layout are gate valves. These two items originally caused such a large frictional head on the pump that it was not able to deliver the load.

Although the average load is only 40 gpm, because of the heavy condensation during starting, required pump capacity is 200 gpm. The piping is 3 in. and the motor size 3 hp. Notwithstanding liberal size of the pump, during the morning and sometimes during the day the returns were heavier than the pump could handle and large amounts of condensate were lost in overflow at the receiving tank. An inspection showed the large number of unnecessary elbows both at the suction and discharge and also the three globe valves where gate valves belonged. Calculations showed that the additional bends and the globe valves were responsible for a frictional head that heavily overloaded the pump and motor, as will be seen from an examination of the following figures:

8 elbows @ 30 pipe diameters each	240	pipe diameters
3 globe valves @ 90 pipe diameters each	270	" "
Total	510	" "
	or 127	ft
Feet of actual piping, pump to heater	40	"
Total effective length	167	"

With a frictional head of 17 ft per 100 ft of piping the total frictional head is 28 ft. (167/100×17).

Since the elevation from the pump to the heater is 15 ft the total head on the pump is 28 + 15 or 43 ft.

*Consulting Engineer, Chicago.

The corresponding calculations for the remodeled layout are as follows:

2 elbows @ 30 pipe diameters each	60	pipe diameters
3 gate valves @ 15 pipe diameters each	45	" "
Total	105	" "
	or 26	ft
Length of actual piping, pump to heater	40	"
Total effective length	66	"

Total frictional head at 17 ft per 100 ft is 11 ft.

This means a total head of only 26 ft as against 43 ft for the original layout. Assuming a 50 per cent efficiency for the pump the amount of water that it can deliver with its 3-hp motor at the two respective heads is:

$$\text{Original layout: } \frac{3 \times 33{,}000\dagger \times 0.50}{43} = 1150 \text{ lb per min} = 145 \text{ gpm}$$

$$\text{Remodeled: } \frac{3 \times 33{,}000 \times 0.50}{26} = 1900 \text{ lb per min} = 240 \text{ gpm}$$

†One horsepower is equal to 33,000 ft lb per minute.

Air Conditioner Saves Its Cost in First Carload of Tomatoes

AN air conditioner installed in the tomato ripening room of the Morris Fruit and Produce Co., Minneapolis, Minn., saved its cost in the first carload of tomatoes by preventing the usual loss from drying out which occurs under ordinary atmospheric conditions.

The ripening room holds two carloads of tomatoes, and before the installation of air conditioning as high as 20 per cent of each carload became spoiled because of loss of moisture. Previous attempts to keep the tomatoes moist by periodic sprinkling failed to meet with appreciable success because only the top layers received the proper amount of moisture.

To promote ripening, the room is maintained at a fairly high temperature, the relative humidity being held at approximately 70 per cent throughout the ripening room. Under these conditions, all tomatoes in the room are not only prevented from drying out, but actually tend to regain their natural moisture content, the fruit remaining firm and solid and commanding better prices.*

*Information from W. T. Cook, General Electric Company, Schenectady, N. Y.

Bureau of Standards finds

Air Conditioning of Libraries

essential to Preservation of Books and Records

THE problem of preserving valuable records, papers, and books has become of increased importance because of the continued concentration of our population in cities. The impurities in the atmosphere polluted by smoke from heating and power plants, exhaust gases from automobiles, gases from industrial processes, etc., exert an extremely deleterious effect on paper which must be counteracted if important records are to be saved for future generations.

Systematic studies over a period of four years on the preservation of valuable records have been conducted by the Bureau of Standards and the more important information which has been secured is summarized in a recent publication.[1]

A survey of material stored in libraries and the conditions surrounding it showed that many valuable publications were badly deteriorated. The main deteriorative agents were indicated to be light, low-grade paper, adverse temperature and humidity, and acidic pollution of the air by sulphuric dioxide. Dust is also a damaging agent, and its removal is recommended. It is thus seen that carefully controlled air conditioning is desirable for libraries, and is highly important where valuable records are stored in urban and industrial localities.

Preventing Damage by Sulphur Dioxide

The efficiency of a conventional air conditioning system with respect to the removal of acid gases not being known, a study was made to learn whether washing the air in the customary manner completely removes sulphur dioxide and, if not, how the air conditioning procedure can be modified to provide air for libraries that is completely free from this harmful constituent. This work was done with the cooperation of the Folger Shakespeare Library in Washington, as its facilities include an air conditioning system for the stacks and vaults.

It had been suggested that the efficiency of existing air conditioning systems with respect to removal of sulphur dioxide might be increased by raising the alkalinity of the wash water. This was accomplished in the course of these experiments by adding a mixture of certain alkaline salts at a rate sufficient to maintain the hydrogen-ion concentration of the wash water within the range pH 8.0 to 9.0.[2] This particular mixture of chemicals forms a passive gelatinous deposit upon metal surfaces, thereby retarding corrosion due to contact with water. In localities where a high degree of atmospheric pollution is encountered, the metal parts of air conditioning systems corrode so rapidly, if preventive measures are not taken, that complete replacement has in certain cases become necessary after less than a year of operation. The hydrogen-ion concentration of the wash water in one system, where rapid deterioration was noted, dropped from pH 7.4 to pH 6.0 in the course of eight hours' operation, because of the absorption of acid from the air washed. It is obvious that protection against corrosion is necessary under such conditions.

Fig. 1 gives the average results of the experimental work for each day from January 6 to February 16, 1933. It compares the sulphur dioxide of the untreated air entering the library with that of the washed air inside, with reference to the effect of washing with untreated water and water treated with the alkaline mixture. This figure also shows the three successive changes or different treatments of the wash water, with the varying hydrogen-ion concentrations which characterized them. From January 6 to 24 the wash water was untreated and unchanged, though enough fresh water was continuously added to make up for that lost by evaporation. During this period the hydrogen-ion concentration of the wash water averaged pH 7. From January 25 to February 4 the water was treated with the alkaline mixture already described at such a rate that the hydrogen-ion concentration of the resulting solution fell only momentarily below pH 8.6. On February 6 the system was drained completely and refilled with fresh, untreated water of pH 7.7.

It is evident from Fig. 1 that air washed with untreated water in an air conditioning system of the customary type, though containing less sulphur dioxide than unwashed air, was still appreciably contaminated. Fig. 1 shows further that complete removal of sulphur dioxide was effected by washing the air with treated water. Although the sulphur dioxide content of the unwashed air varied from hour to hour and from day to day, the amount of sulphur dioxide present in the air after washing apparently depends upon the hydrogen-ion concen-

[1] "Summary Report of Bureau of Standards Research on Preservation of Records," by A. E. Kimberly and B. W. Scribner, *Bureau of Standards Miscellaneous Publication* No. 144. Available from the Superintendent of Documents, Washington, D. C. Price, 5c.

[2] Hydrogen-ion concentration is measured by the pH scale, pH 7.0 being neutral. Values below pH 7.0 are acidic, above are alkaline.

tration of the wash water. This relationship is illustrated in Fig. 2. As the alkalinity of the wash water decreased because of the absorption of sulphur dioxide coupled with insufficient additions of the alkaline material, the sulphur dioxide content of the washed air rose. Air treated with water having a hydrogen-ion concentration of pH 8.6 or above was completely free from sulphur dioxide and, in that respect, suited to use in libraries.

It is therefore recommended that libraries, particularly in localities where a high degree of atmospheric pollution is encountered, be equipped with air washers; or better; complete air conditioning systems. To insure complete removal of sulphur dioxide from the air, the water in the air washer should be changed at least once a week and treated continuously with an alkaline solution so that the hydrogen-ion concentration of the wash water may always fall within the range pH 8.5 to 9.0. All air returning from the storage spaces should be rewashed to eliminate the chance of recirculating any sul-

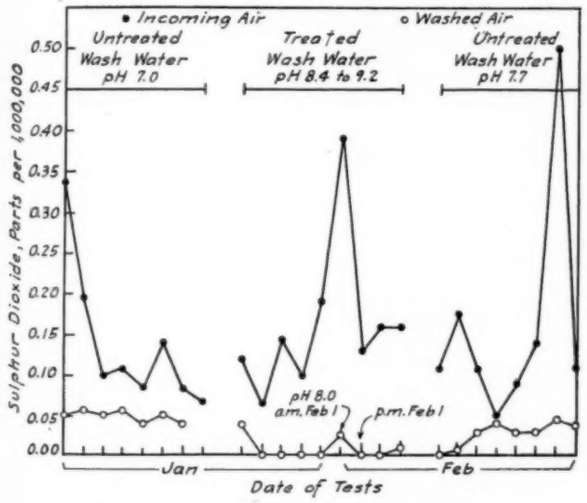


Fig. 1—Comparison of the effect of washing with untreated and treated water on the sulphur dioxide content of library air

phur dioxide that may have leaked into the system. It is considered best to run the entire air conditioning apparatus continuously.

Importance of Temperature and Humidity

Paper, cloth, thread, adhesives, and leather, the organic constituents of books, are also susceptible to decay from chemical changes brought about by deteriorative components of the book materials, and sunlight. There is also the rotting effect of mildew which results from spores carried by the air. All of these deteriorative actions are stimulated by certain temperatures and humidities.

Book paper saturated with water loses nearly all its strength; but it is not conceivable that the paper in a book could ever pick up this amount of water from the air. Perfectly dry paper is relatively brittle. Prolonged exposure of a book to an atmosphere of less than 15 per cent relative humidity might render the pages tender. Direct or prolonged heating will also make paper brittle. so books should not be placed near steam pipes or radiators.

TO PARAPHRASE, "stone walls do not a library make"—the preservation of records is the chief function of a library. Preservation involves protection against harmful effects of temperature and humidity, acidic gases in the atmosphere, light, dust, etc. As the result of four years of research the Bureau of Standards recommends, in the report summarized here, carefully controlled air conditioning, treatment of the air washer water with an alkaline solution, removal of dust. Suggestive of the modernization of library buildings with air conditioning are these recommendations

The cloth and thread, whether of cotton or linen, are not materially affected by any extremes of humidity or temperature which could normally be expected in a library. The thread in much-used books sometimes gives way to a repeated stress well within its ultimate strength. The cloth may be sized with starch or similar water soluble adhesive. This may become sticky if the humidity is too high, or brittle if it is too low, and is susceptible to attack by oxygen in the presence of moisture and light.

Glue and casein are colloidal materials containing, normally, much water. If the humidity gets too high (much

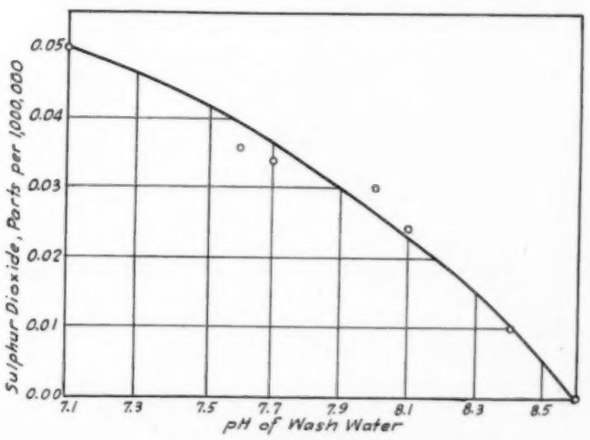


Fig. 2—Relationship of the sulphur dioxide content of washed air to the hydrogen-ion concentration of the wash water

above 70 per cent) they are likely to become sticky, which probably does no harm. But if the humidity drops much below 40 per cent, glue which is stored in contact with such an atmosphere for a considerable length of time is apt to become so brittle that the sudden opening of the book will break it. Glue is permanently damaged by prolonged exposure to temperatures below freezing (25 F may be taken as the freezing point of glue containing a little water).

Leather is not affected by temperature or humidity within the ordinary ranges. The oils and greases which

are worked into leather to preserve its flexibility have a definite, though very low, vapor pressure. They are gradually lost by evaporation and should be replaced at intervals by rubbing the leather with neat's-foot or similar oil.

Mildew is a term popularly used to designate any one of a variety of vegetable organisms. All of these have certain characteristics in common; they propagate by means of spores, which are almost always present in the air. They can remain dormant for an indefinite period until the conditions become right for their growth. They grow very slowly at low temperatures (40 F), very rapidly at the optimum temperature (which differs with the species), and are killed by high temperatures (some species as low as 100 F). The different species subsist on different kinds of food; many kinds can live on glue and starch, and at least one is capable of digesting the pure cellulose found in cloth and paper. They all require an abundance of water. Unless the growth of mildew continues unchecked for a considerable time, it is not likely to do any harm other than create an unsightly appearance. If the relative humidity is kept under 80 per cent, paper, cotton, or leather will not contain sufficient water to support the growth of mildew; glue and starch will. These latter materials can be treated to prevent mildew growth, beta-naphthol being generally used for this purpose.

Dust may be composed partly of the product of the slow growth of mildew. It consists mostly, however, of small angular particles of siliceous material. These particles are so small that they can become embedded between the fibers of the leather or the cloth, and perhaps even of the paper. Subsequent flexing of the material may cause the angular dust particles to cut the fibers. The danger of damage from dust is so slight that it can be eliminated by a reasonable amount of care.

It is necessary to consider how the relative humidity in a building varies from time to time. During the summer, if the building is neither heated nor cooled, the temperature and relative humidity of the air inside the building and out-of-doors will on the average be the same, and the relative humidity prevailing under these conditions is in most cases not injurious to the books. It is true that if a period of relatively cool weather is followed by a period of warm damp weather, the warm nearly-saturated air coming in contact with the cool books, may deposit moisture on them or increase the moisture content to a point where mildew may develop. In winter the conditions are different. The outside air has a rather high relative humidity, but being cold, its moisture content is small. When this air is heated, its moisture content is not changed appreciably, but while it is warm, its relative humidity is much lower than that of the outside air, and as it passes over the books it dries them out in some cases sufficiently to injure them. Hot air is not necessarily dry air, but if cold air is heated without adding moisture to it, its relative humidity is decreased and it becomes "dry." To maintain a higher relative humidity in a heated building in winter it is necessary to add water vapor as the air is heated.

In general, then, the air in a heated building has a rather high relative humidity in summer and a low relative humidity in winter, even though the relative humidity of the outside air may be nearly the same in both cases. Consequently, during the winter the books dry out, while in summer the moisture content increases in some cases to an extent sufficient to cause injury.

In considering conditions of temperature and relative humidity for the storage of records, economy of operating expense and comfort of the personnel caring for the records must be considered as well as the preservation of the records, although this must be the determining factor. It is, therefore, recommended that 70 F and 45 per cent relative humidity be maintained in storage spaces in the winter and 85 F and 45 per cent relative humidity be maintained during the summer months. It is felt that these conditions are satisfactory for the preservation of records, and the higher temperature recommended for the summer months will result in lower operation costs as well as lessen the shock to employees upon leaving or entering the conditioned area. Under conditions such as these, the stored material possesses good strength and flexibility, and the growth of molds and fungi is minimized.

A positive circulation and diffusion of air through the library, or at least the storage stacks, is desirable, in that it will facilitate proper control of temperature and humidity conditions. It is necessary to take in a supply of outside air only to meet the requirements of the people in the library. Temperature, humidity, and chemical conditions of the air can be maintained at their optimum values (so far as the books themselves are concerned) at lowest cost and with greatest ease, by recircu'ation of the air within the building. Dust should be removed from the air.

Method of Installing the Panel Heating System in British Embassy

Have you any information on the heating and roof construction of the British Embassy in Washington? A friend informs me the roof insulation is corkboard imbedded between two separate courses of concrete, heating pipes running through the lower concrete which is plastered and heat reflected down into the room.—W. S. H.

Answer: There is no corkboard between two courses of concrete in this job. As far as the heating is concerned however, it is correct, that hot water coils were imbedded in the bottom of the concrete slab. This was done to install the panel heating system of radiant heating, which was used at the British Embassy.

The panel heating system consists of warming radiant panels by means of coils imbedded in the construction and in direct contact with the radiant surfaces, which are usually plastered ceilings. The construction used in the British Embassy consisted of I-beams with the bottom arch construction and the coils were imbedded in the bottom of the arch. No insulation was used in this case. In certain parts of the Embassy and in subsequent panel systems installed in this country, the coils were hung at the hung ceiling level, and the wire laths tied right to the coil pipes, and the plaster applied to the wire lath. In this case we used 2-in. corkboard as an insulation above the pipes to prevent loss of heat in the hung ceiling space.

Panel heating depends on the emission of radiant heat from the warm panels which heats the occupants of the room as well as the contents. The air is heated by secondary convection. The air temperature need only be approximately 63 F to correspond to 70 F in a convected heated room.—Leon L. Munier*.

*Wolff & Munier, Inc., U. S. Licensees of Richard Crittall & Co., Ltd., panel heating system, New York City.

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Factors Affecting the Heat Output of Convectors

By A. P. Kratz*, M. K. Fahnestock** and E. L. Broderick,*** *(MEMBERS)*

Urbana, Ill.

***This paper is the result of research conducted at the University of Illinois in cooperation with the Research Laboratory of the* AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.**

The data presented in this paper were obtained in connection with an investigation carried on in the Department of Mechanical Engineering as part of the work conducted by the Engineering Experiment Station of the University of Illinois. The investigation was conducted under the direction of A. C. Willard, Acting Director of the Engineering Experiment Station and Head of the Department of Mechanical Engineering. This paper includes results from the work constituting a continuation of the study of the performance of convectors and the material will ultimately comprise part of a bulletin of the Engineering Experiment Station.

TWO previous papers[1,2] on this subject were devoted to discussions of tests run in a warm wall testing booth for the purpose of deciding the validity of the correction factor, applied in the A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[3] for reducing the heat output obtained under test conditions to the equivalent heat output under standard conditions with steam in the heating unit at 215 F and a temperature of 65 F for the air at inlet. This correction factor is:

$$C = \left[\frac{150}{t_s - t_1}\right]^{1.3} \qquad (1)$$

where 150 = the temperature difference between steam at 215 F and inlet air at 65 F

t_s = the temperature of steam during the test, degrees Fahrenheit

t_1 = the average inlet air temperature during the test, degrees Fahrenheit

The results[1,2] in general indicated a reasonably close agreement between the heat output actually obtained with steam at 215 F and inlet air at 65 F and that calculated from the heat output at some other inlet air temperature through the medium of the correction factor, provided that the inlet air temperature did not exceed 80 F or fall below 60 F. Deviations as great as 4.5 per cent were observed, however, and it was recommended that the testing range be confined to inlet air temperatures between the limits of 60 F and 75 F. There seemed to be some indication that the percentage deviation between the actual and calculated heat outputs might be affected by the physical dimensions of the convectors, particularly by the height of the cabinet.

No analysis was made to determine whether the variation in heat output with the difference in temperature between the steam and inlet air for all of the types of convectors tested could be represented by an equation of the form:

$$H = K \quad (t_s - t_1)^n \qquad (2)$$

where H = heat output, Btu per hour

K = a constant

t_s = the temperature of the steam, degrees Fahrenheit

t_1 = the inlet air temperature, degrees Fahrenheit

n = a numerical exponent

and whether the correction formula should assume the form:

$$_cH_{150} = {}_aH_{(t_s - t_1)}\left[\frac{150}{t_s - t_1}\right]^n \qquad (3)$$

where $_cH_{150}$ = the calculated, or corrected heat output at the standard temperature difference with steam at 215 F and inlet air at 65 F, Btu per hour

$_aH_{(t_s - t_1)}$ = the actual heat output at the observed temperature difference $(t_s - t_1)$, Btu per hour

t_s = the observed steam temperature, degrees Fahrenheit

t_1 = the observed inlet air temperature, degrees Fahrenheit

n = a numerical exponent

The tests in the warm wall booth were therefore continued for the purpose of consolidating all of the results, and determining whether the heat outputs of all of the types of convectors tested could be represented by an equation of the form of equation (2); and whether some exponent, n, differing from 1.3 used in the correction formula (3) would be more representative of all of the types of convectors tested and produce less individual deviations between the observed and corrected values of the heat outputs at the standard temperature difference of 150 F than the value 1.3 recommended in the Standard Code for Rating and Testing Concealed Gravity Type Radiation (Steam Code). Further tests were also run for the purpose of making a more extensive study of the effect of the height of the cabinet on the heat output and the value of n for two types of convectors.

A previous paper[4] discussed the use of the eupatheoscope for evaluating a complex or non-uniform environment in terms of the equivalent temperature of a uniform environment that could be correlated with human comfort. The principle underlying this instrument involves the measurement of the heat loss from a sizable body when the surface is maintained at a constant temperature. The calibration is based on measurements of the heat loss from the eupatheoscope at different air temperatures when both the surrounding walls and the air are maintained at the same temperature. The later use

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of the instrument implies that for a given uniform environment with walls and air at the same temperature no appreciable change in the heat loss occurs with variations in the relative humidity of the surrounding air. A

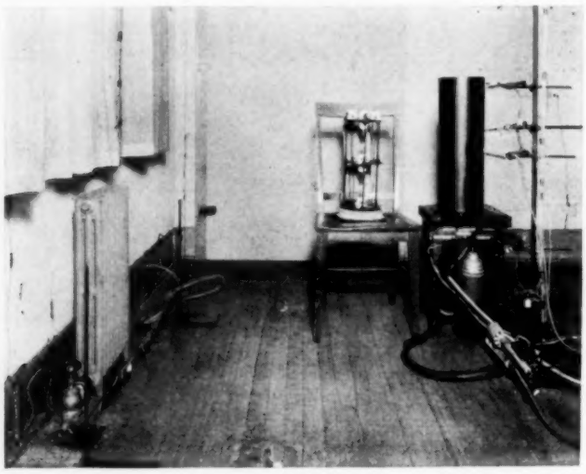

Fig. 1—The eupatheoscope with heating element removed

study was therefore made in the low temperature testing plant to determine the effect of the relative humidity of the surrounding air on the heat loss from the eupatheoscope. This study was then extended to include the determination of the effect of the relative humidity of the air on the steam condensation or heat output obtained with one type of convector having a non-ferrous heating unit.

Description of Apparatus

Both the warm wall testing booth[1,2,3] and the low temperature testing plant [1,4,5,6,7] have been completely described in previous publications.

The eupatheoscope, shown in Fig. 1, consisted of a hollow copper cylinder 7½ in. in diameter and 22 in. long, placed vertically on a stool so that its mean height was 30 in. above the floor. Thirteen thermocouples, made from No. 34 B.&S. gage copper and constantan wire were embedded flush with the surface of the cylinder, which was blackened. These 13 thermocouples were electrically but not thermally insulated from the cylinder and were connected in series to a recording-controlling potentiometer, thus affording a means of both indicating and controlling the average temperature of the surface of the cylinder.

In order to heat the surface, current from the line was supplied to a lamp-bank placed inside of the cylinder. This lamp-bank, removed from the cylinder, is shown in Fig. 1. The lamps, arranged as shown, in order to give uniform surface temperature were connected in two separate circuits. Parts of the lamps were supplied with a constant amount of current just insufficient to maintain the surface at the required temperature. Current supplied to the rest of the lamps served as an accurate control to maintain the surface at the required temperature, and the total current was measured by means of a sensitive wattmeter.

In all, 7 different types of convectors and an 8-sect., 26-in., 5-tube cast-iron radiator were tested in the warm wall booth. Four of the different types of convectors had non-ferrous heating units, two had cast-iron heating units, and one had a bimetallic heating unit. Two of the non-ferrous heating units were tested with 5 different heights of cabinets and one cast-iron unit was tested with 2 heights of cabinets. One type of non-ferrous convector was tested with two lengths of heating units, and two types of non-ferrous convectors and one type of cast-iron convector were tested with two different widths of heating units. The dimensions of the different combinations are shown in the diagrams inserted in Figs. 2 to 8 inclusive and in Table 1.

Test Procedure

All tests were run with a steam temperature of 216.5 F in the convectors. In the case of the warm wall booth, curves were obtained, establishing the relation between the heat output, as measured by steam condensation, and the temperature of the air at inlet, by conducting tests on each convector at different inlet air temperatures varying over a range of from 60 F to 90 F. In order to accomplish this, the large room in which the test booth was erected was heated or cooled to a temperature approximating the desired inlet air temperature, and the convector was allowed to establish the temperature conditions in the booth necessary for thermal equilibrium.

For some of the earlier tests in the warm wall booth the thermometers for obtaining the temperature of the air at inlet were not shielded against radiation. For later tests these thermometers were shielded in order to obtain the true air temperature. In the cases of the usual types of convectors, equipped with separate cabinets, the correction for radiation to the inlet thermometers did not

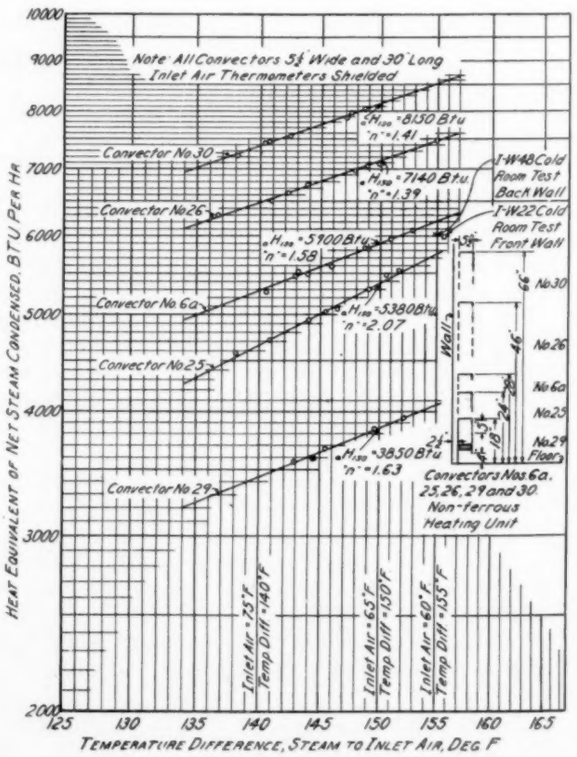
Note: All Convectors 5½" Wide and 30" Long
Inlet Air Thermometers Shielded
Convector No 30
Convector No 26
Convector No 6a
Convector No 25
Convector No 29
I-W48 Cold Room Test Back Wall
I-W22 Cold Room Test Front Wall
Wall
No 30
No 26
No 6a
No 25
No 29
Floor
Convectors Nos 6a, 25, 26, 29 and 30 Non-ferrous Heating Unit
Inlet Air = 75°F Temp Diff = 140°F
Inlet Air = 65°F Temp Diff = 130°F
Inlet Air = 60°F Temp Diff = 135°F
HEAT EQUIVALENT OF NET STEAM CONDENSED, BTU PER HR
TEMPERATURE DIFFERENCE, STEAM TO INLET AIR, DEG. F

Fig. 2—Performance curves for convectors Nos. 6a, 25, 26, 29 and 30 showing effect of height of cabinet

exceed approximately one per cent, and hence any comparisons made from the results would not be affected. In the case of special types, in which the whole front was radiating surface in contact with steam, a large correction would be necessary, and the tests on these types were all run with shielded thermometers. In all cases the curve sheets indicate whether or not the thermometers were shielded.

In the few cases in which the convector was also tested in the low temperature testing plant, the temperature in the cold room was maintained at about —2.0 F, and one of the exposed walls was subjected to an equivalent wind velocity of approximately 10 mph. The temperature above the ceiling was maintained at 62 F and the air in the space below the floor at such a temperature that the upper surface of the floor was approximately 2 F

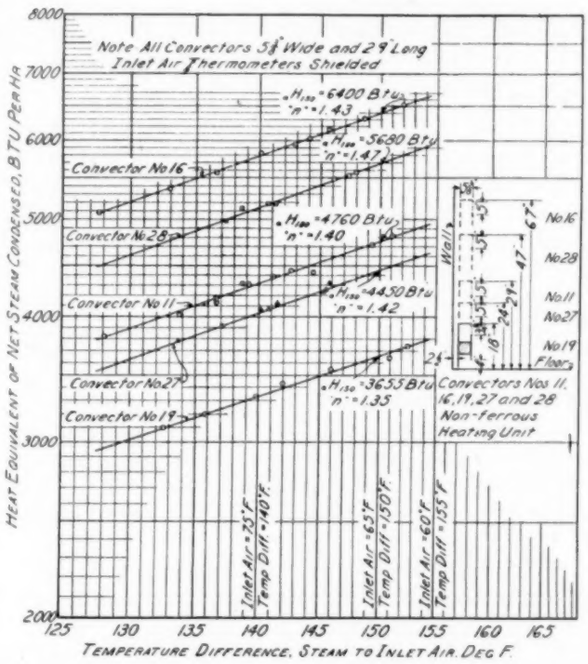
Note All Convectors 5¾" Wide and 29" Long
Inlet Air Thermometers Shielded
Convector No 16
Convector No 28
Convector No 11
Convector No 27
Convector No 19
Convectors Nos 11, 16, 19, 27 and 28 Non-ferrous Heating Unit
Heat Equivalent or Net Steam Condensed, B.T.U. Per Hr.
Temperature Difference, Steam to Inlet Air, Deg. F.

Fig. 3—Performance curves for convectors Nos. 11, 16, 19, 27 and 28 showing effect of height of cabinet

warmer than the lower surface. Each convector was allowed to establish whatever temperature conditions were necessary in the room in order to maintain equilibrium between the heat loss from the room and the heat output of the particular convector. In selecting the sizes of the convectors, however, the selection was limited to sizes that would not either overheat or underheat the room an unreasonable amount; that is, temperatures above 75 F or below 65 F at the 30-in. level were not accepted.

For the tests run to determine the effect of relative humidity on the steam condensation of a convector the temperatures in the cold room and in the spaces above the ceiling and below the floor were varied in order to obtain a range of inlet air temperatures of from 60 F to 70 F when the convector was allowed to establish conditions of temperature equilibrium in the room. A curve of heat output was thus first established with the relative humidities normally obtained without the addition of any water vapor in the room; or with relative humidi-

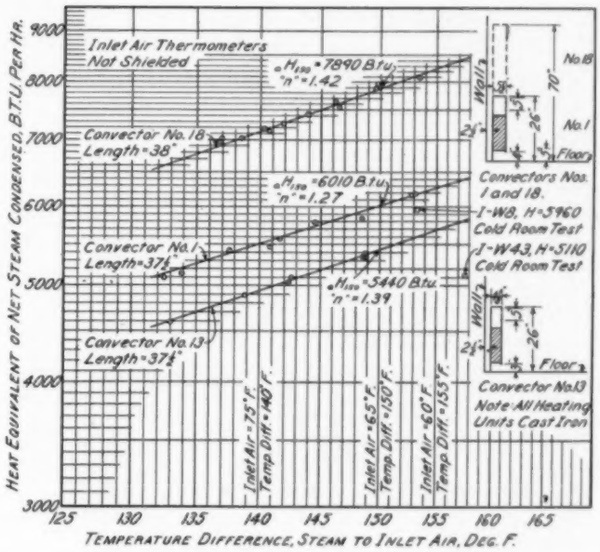
Inlet Air Thermometers Not Shielded
Convector No. 18 Length = 38"
Convector No. 1 Length = 37½"
Convector No. 13 Length = 37½"
Convectors Nos. 1 and 18.
1-W8, H=5960 Cold Room Test
1-W43, H=5110 Cold Room Test
Convector No. 13 Note: All Heating Units Cast Iron
Heat Equivalent of Net Steam Condensed, B.T.U. Per Hr.
Temperature Difference, Steam to Inlet Air, Deg. F.

Fig. 4—Performance curves for convectors Nos. 1, 13 and 18 showing effect of height of cabinet

ties ranging from 6 to 15 per cent. Water vapor was then added to the air in the room by evaporating water in an open tank by means of a bayonet type of electrical heater. As the humidity was increased, the convector was allowed to establish temperature equilibrium and the steam condensation was obtained for a number of relative humidities, including a maximum of 68 per cent.

In both the test room and the test booth, no test observations were made until conditions had remained constant for several hours, as indicated by readings of all thermocouples, or thermometers. When the required thermal constancy had been attained, the condensate was weighed over a period of one hour, and no test was accepted if the condensate showed more than 2½ per cent deviation in the successive 10-min. increments of weight. At the end of each test, a separate test was made to determine the condensation in the piping alone, and the total condensate was corrected by subtracting the amount so determined.

In determining the effect of relative humidity on the heat loss from the eupatheoscope a curve establishing the relation between heat loss and equivalent temperature at a normal relative humidity of 30 per cent, and with the surface temperature of the eupatheoscope at 83 F was first obtained. This was done by placing the instrument in the test room of the low temperature testing plant, and allowing the room to come to temperature equilibrium with the air and the inside surfaces of the walls at the same temperature for a number of different temperatures. No form of heating unit was used in the room for this purpose, but the different equilibrium temperatures were obtained by adjusting the temperature of the air in the laboratory in which the low temperature testing plant was located and then closing the test room and making the observations when equilibrium was established. The relative humidity in the test room was then varied over a range of from 25 to 80 per cent by means of water sprays and an atomizing humidifier in the cold room and the observations were repeated at several different temperatures, in every case with the wall surfaces and air at the same temperature.

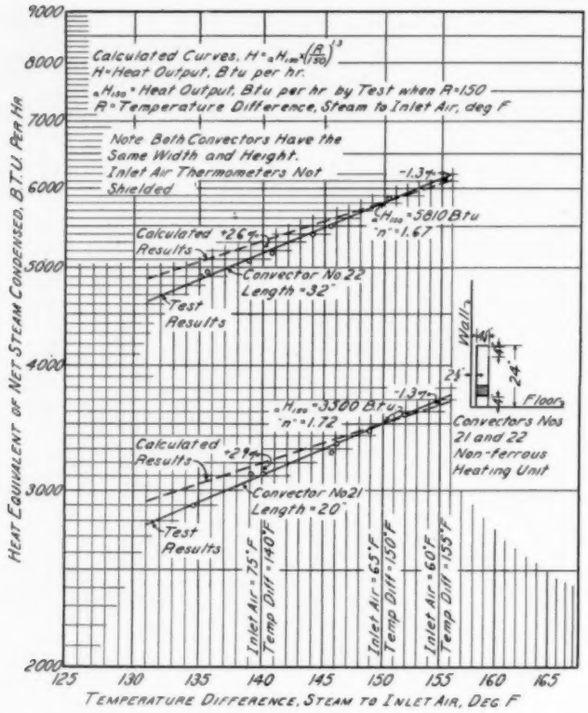
Calculated Curves, H = aH150 × (R/150)^n
H = Heat Output, Btu per hr.
aH150 = Heat Output, Btu per hr by Test when R = 150
R = Temperature Difference, Steam to Inlet Air, deg F
Note Both Convectors Have the Same Width and Height.
Inlet Air Thermometers Not Shielded
aH150 = 5810 Btu
n = 1.67
Calculated Results
Test Results
Convector No 22
Length = 32"
aH150 = 3300 Btu
n = 1.72
Convector No 21
Length = 20"
Walls
Floor
Convectors Nos 21 and 22 Non-ferrous Heating Unit
Inlet Air = 75°F
Temp Diff = 140°F
Inlet Air = 65°F
Temp Diff = 150°F
Inlet Air = 60°F
Temp Diff = 155°F
Heat Equivalent of Net Steam Condensed, B.T.U. Per Hr
Temperature Difference, Steam to Inlet Air, Deg F

Fig. 5—Performance curves for convectors Nos. 21 and 22 showing effect of length

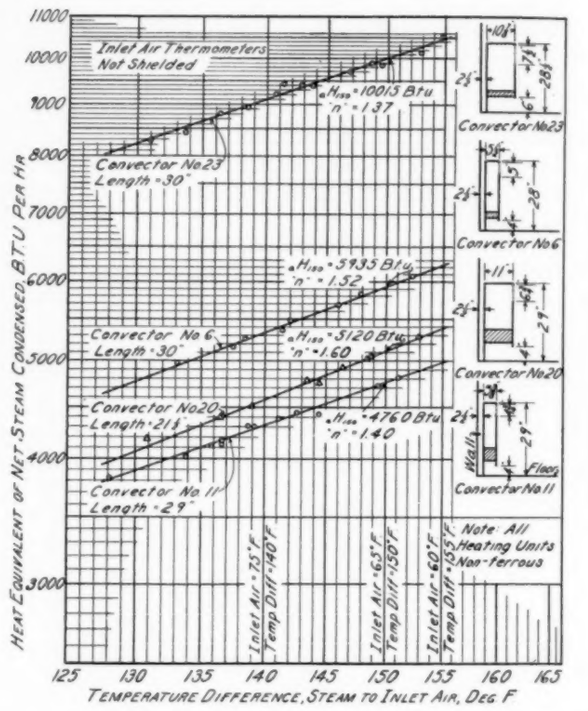

Fig. 6—Performance curves for convectors Nos. 6 and 23 and Nos. 11 and 20 showing effect of width

Results of Performance Tests

The results of all of the performance tests conducted on the convectors in the warm wall booth have been plotted on logarithmic coordinates and are shown in Figs. 2 to 8 inclusive. In Figs. 2 to 6 the tests have been classified by groups showing the effects of heights of cabinets and of length and width of the heating units. Figs. 7 and 8 consist of two groups of convectors not subject to any particular classification, but which were included in the general study of the performance of different types. Fig. 7 also includes results obtained with the 8-sec. 26-in. 5-tube direct cast-iron radiator.

Inspection of the curves in Figs. 2 to 8 inclusive indicates that in every case the data can best be represented by a straight line including the whole range of test results. It is therefore evident that for all of the convectors tested, the relation between the heat output and the difference in temperature between the steam and the inlet air can be expressed by a general equation of the form of equation (2) in which n and K assume different numerical values depending on the particular type of heating unit or combination of cabinet and heating unit, involved.

Since the performance curves for all of the convectors tested are of the form expressed by equation (2), it is further evident that the formula for correcting the observed heat output under test conditions to an equivalent heat output under standard conditions should assume the form of equation (3); and that the value of n for each particular convector may be determined from the slope of the straight line representing the test data when the latter are plotted on logarithmic coordinates. These values of n have been indicated on the test curves in Figs. 2 to 8. It should be noted in this connection that the temperature difference scale employed in Figs. 2 to 8 is four times the heat output scale, and therefore in order to obtain a graphical solution for the slopes of the different lines, the measured vertical distance should be multiplied by 4.

The present practice in correcting test results as recommended in the A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[3] is to employ a value of $n = 1.3$ in equation (3) for all types of convectors. For the purpose of comparison, curves having a slope of 1.3 have been shown for convectors Nos. 21 and 22 in Fig. 5. These curves have been made coincident with the actual test curves at points corresponding to the standard temperature difference of 150 F between steam and inlet air. The deviations between the theoretical curves and the actual test curves at various temperature differences are representative of the errors[1] that would result if the observed heat outputs at these temperature differences were corrected to the equivalent heat outputs at the standard temperature difference of 150 F. It may be observed that the error becomes greater as the allowable limits for the temperature difference obtained on actual tests deviate further from the standard temperature difference of 150 F. The present practice is to confine the allowable testing range to limits between temperature differences of 140 F and 155 F; or to inlet air temperatures between 75 F and 60 F. It is evident that any method of correction that is acceptable with inlet air at 75 F will also be acceptable for inlet air at 60 F. Hence, this study has been confined to an analysis of the errors resulting when heat outputs observed with inlet air temperature at 75 F are corrected to equivalent heat outputs at the standard inlet air temperature of 65 F with steam at 215 F.

A study of the actual values of n shown by the test data in Figs. 2 to 8, and listed in Table 1, indicates that

Table 1—Deviation of Calculated Correction from Test Curve

Fig. No.	Convector No.	Type of Heating Unit	Overall Dimensions of Convector, Inches: Length	Height	Width	Height of Heating Unit, Inches	Actual (n) from Slope of Test Curve	Heat Output from Test Curve Btu per Hour: 75 F Inlet aH_{140}	Standard 150 F Diff. or 65 F Inlet, aH_{150}	Heat Output at 75 F Inlet Corrected to Standard 65 F Inlet: By n = 1.3 as per Code: Btu per Hour cH_{150}	Percent Diff. from Test Curve	By n = 1.5, Actual Ave.: Btu per Hour cH_{150}	Percent Diff. from Test Curve
1	2	3	4	5	6	7	8	9	10	11	12	13	14
2	29	Non-ferrous	30	18	5½	2	1.63	3440	3850	3765	−2.20	3820	−0.78
2	25	Non-ferrous	30	24	5½	2	2.07	4660	5380	5100	−5.21	5170	−3.90
2	6a	Non-ferrous	30	28	5½	2	1.58	5290	5900	5790	−1.87	5865	−0.59
2	26	Non-ferrous	30	46	5½	2	1.39	6490	7140	7105	−0.49	7200	−0.84
2	30	Non-ferrous	30	66	5½	2	1.41	7390	8150	8090	−0.74	8200	+0.61
3	19	Non-ferrous	29	18	5⅜	5	1.35	3330	3655	3645	−0.27	3690	+0.96
3	27	Non-ferrous	29	24	5⅜	5	1.42	4040	4450	4420	−0.67	4480	+0.67
3, 6	11	Non-ferrous	29	29	5⅜	5	1.40	4320	4760	4730	−0.63	4790	+0.63
3	28	Non-ferrous	29	47	5⅜	5	1.47	5140	5680	5625	−0.97	5700	+0.35
3	16	Non-ferrous	29	67	5⅜	5	1.43	5790	6400	6340	−0.94	6420	+0.31
4	13	Cast Iron	37½	26	4¼	14⅜	1.39	4940	5440	5410	−0.55	5480	+0.73
4	1	Cast Iron	37½	26	5½	14⅜	1.27	5500	6010	6020	+0.17	6100	+1.50
4	18	Cast Iron	38	70	5½	14⅜	1.42	7140	7890	7820	−0.89	7920	+0.38
5	21	Non-ferrous	20	24	4¼	4	1.72	3105	3500	3400	−2.86	3440	−1.72
5	22	Non-ferrous	32	24	4¼	4	1.67	5170	5810	5660	−2.58	5735	−1.29
6	20	Non-ferrous	21½	29	11	5	1.60	4575	5120	5010	−2.15	5075	−0.88
6	6	Non-ferrous	30	28	5½	2	1.52	5340	5935	5850	−1.43	5920	−0.25
6	23	Non-ferrous	30	28½	10⅞	2	1.37	9110	10015	9980	−0.35	10100	+0.85
7	3	Non-ferrous	21	30⅜	7½	6	1.36	4540	4975	4970	−0.11	5035	+1.21
7	9RC	Cast Iron	20	20	7½	20	1.42	4810	5300	5260	−0.75	5335	+0.66
7	2	Non-ferrous	31	30⅜	5	6	1.35	5080	5575	5560	−0.27	5635	+1.08
7	8R	Cast Iron	20	26	8	26	1.30	5975	6535	6535	0.00	.....	.....
8	5	Non-ferrous	36	18	5½	2	1.72	4680	5285	5120	−3.18	5190	−1.80
8	7	Bimetallic #	22⅝	28⅜	6⅛	20	1.30	5780	6330	6330	0.00	6410	−1.26
8	17	Non-ferrous	26	66	5½	2	1.37	6000	6600	6570	−0.45	6655	+0.83
Average †							1.50				1.24		1.00

†The 8-sect., 26-in., 5-tube, C. I. Radiator, No. 8R, was excluded in obtaining these averages and the percentage differences were averaged without regarding the sign.

they vary from a minimum of 1.27 to a maximum of 2.07, but that, excluding the 8-sec., 26-in., 5-tube direct cast iron radiator, 22 values out of a total of 24 were greater than 1.3. Hence, it appears reasonable to assume that a value of n greater than 1.3 should be used in the correction formula applying to all types of convectors if the error is to be reduced to a minimum. The average value for n for all of the convectors tested was approximately 1.5. In Table 1 the equivalent heat outputs at the standard temperature difference of 150 F have been calculated from the actual heat outputs observed at an inlet air temperature of 75 F, as read from the curves at a temperature difference of 140 F, using the values of both 1.3 and 1.5 for n in the correction formula given in equation (3). These calculated values have been compared with the actual heat outputs read from the curves at a temperature difference of 150 F, and the percentage difference between the calculated and the actual heat outputs has been listed in columns 12 and 14. A percentage having a plus sign indicates that the calculated value was greater than the actual.

From column 12 it may be observed that when a cor-

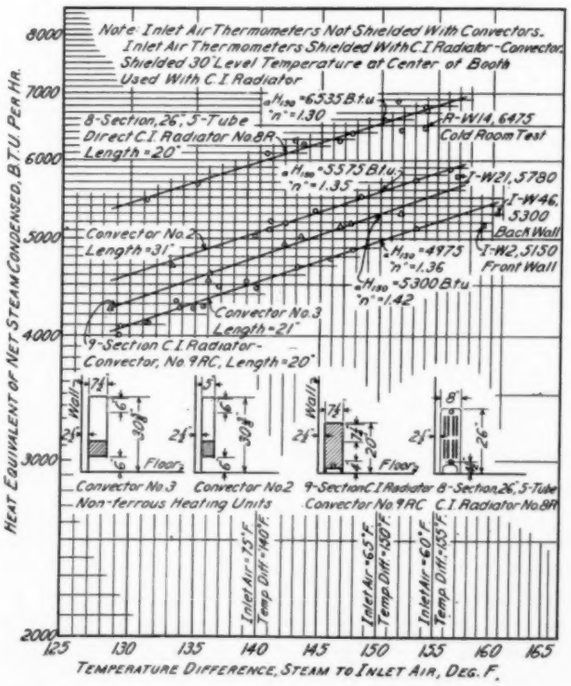
Note: Inlet Air Thermometers Not Shielded With Convectors.
Inlet Air Thermometers Shielded With C.I. Radiator-Convector Shielded 30" Level Temperature at Center of Booth Used With C.I. Radiator
8-Section, 26", 5-Tube Direct C.I. Radiator No. 8R
Length = 20"
Convector No. 2
Length = 31"
Convector No. 3
Length = 21"
9-Section C.I. Radiator-Convector, No. 9RC, Length = 20"
Cold Room Test
Back Wall
Front Wall
Floor
Convector No. 3 Non-ferrous Heating Units
Convector No. 2
9-Section C.I. Radiator Convector No. 9RC
8-Section, 26", 5-Tube C.I. Radiator No. 8R
Heat Equivalent of Net Steam Condensed, Btu. Per Hr.
Temperature Difference, Steam to Inlet Air, Deg. F.

Fig. 7—Performance curves for convectors Nos. 2 and 3, C. I. radiator-convector, and direct C. I. radiator

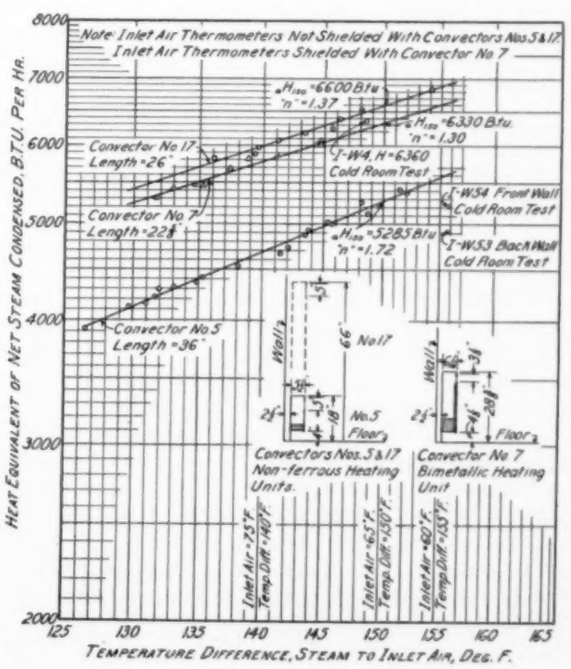
Note: Inlet Air Thermometers Not Shielded With Convectors Nos. 5 & 17
Inlet Air Thermometers Shielded With Convector No. 7
Convector No. 17
Length = 26"
Convector No. 7
Length = 22½"
Convector No. 5
Length = 36"
Cold Room Test
Front Wall
Back Wall
Floor
Convectors Nos. 5 & 17 Non-ferrous Heating Units
Convector No. 7 Bimetallic Heating Unit
Heat Equivalent of Net Steam Condensed, Btu. Per Hr.
Temperature Difference, Steam to Inlet Air, Deg. F.

Fig. 8—Performance curves for convectors Nos. 5, 7, and 17

rection factor with a value of $n = 1.3$ was used, the equivalent heat output at 150 F temperature difference was, with one exception, less than the actual, and the average error, disregarding the sign, was 1.24 per cent. In the case of column 14 for which a value of $n = 1.5$ was used the equivalent heat output was sometimes greater and sometimes less than the actual, and the average error, disregarding the sign was 1.00 per cent. For $n = 1.3$ there were 7 cases in which the error was greater than 1.5 per cent, and 6 cases in which it was greater than 2 per cent, while for $n = 1.5$ there were only 4 cases in which the error was greater than 1.5 per cent and only one case in which it was greater than 2 per cent. Furthermore, comparing the individual values in columns 12 and 14, it may be observed that in 15 cases out of the 24, excluding the cast-iron direct radiator, the numerical value of the percentage error was reduced when $n = 1.5$ was used, while in 9 cases the numerical value of the percentage error was increased. Hence, from the evidence presented, it seems reasonable to recommend that for convectors a correction factor:

$$C = \left[\frac{150}{t_s - t_1}\right]^{1.5}$$

be employed in the A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) instead of the correction factor:

$$C = \left[\frac{150}{t_s - t_1}\right]^{1.3}$$

now in use.

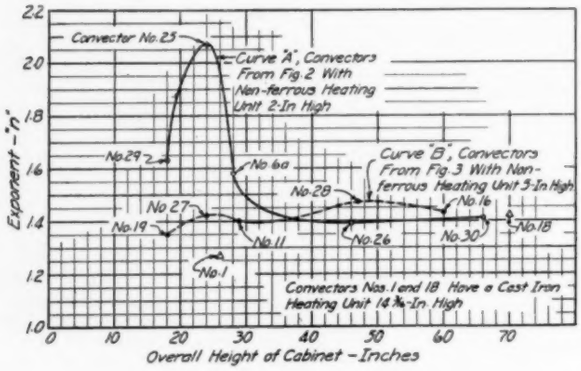
Exponent - "n"
Overall Height of Cabinet - Inches
Convector No. 23
Curve "A", Convectors From Fig. 2 With Non-ferrous Heating Unit 2-In High
Curve "B", Convectors From Fig. 3 With Non-ferrous Heating Unit 3-In High
No. 29
No. 6a
No. 28
No. 16
No. 27
No. 19
No. 11
No. 26
No. 30
No. 18
No. 1
Convectors Nos. 1 and 18 Have a Cast Iron Heating Unit 14 3/4-In. High

Fig. 9—Relation between exponent n and height of cabinet for three types of convectors

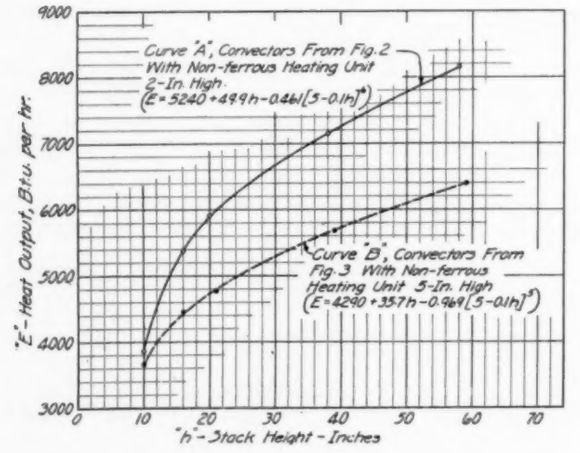
"E" - Heat Output, Btu. per hr.
"h" - Stack Height - Inches
Curve "A", Convectors From Fig. 2 With Non-ferrous Heating Unit 2-In. High (E = 5240 + 499h - 0.46l[5 - 0.1h])
Curve "B", Convectors From Fig. 3 With Non-ferrous Heating Unit 3-In. High (E = 4240 + 357h - 0.969[5 - 0.1h])

Fig. 10—Relation between heat output and stack height for two types of convectors

Note: The equations for curves *A* and *B* in Fig. 10 are intended to cover only the practical ratings within the range of those on which tests have been made.

Since radiation constitutes a very small portion of the heat transfer in the case of convectors, it would seem reasonable to expect that the value of n should approach 1.25 if the heat transfer occurred by free convection. The fact that the values of n are practically all greater than 1.35 for convectors may indicate that the stack effect of the cabinet definitely places the heat transfer in the realm of forced convection. In this case the heat transfer will be a function of the velocity of the air through the heating unit, which will depend on relations existing between the frictional resistance through the unit, the stack height, and the temperature of the air in the cabinet; and very little consistency may be expected in the values of n for different types of heating units and different combinations of cabinets and heating units.

In general, types of convectors having performance curves with higher values of n are advantageous in that they represent types in which the heat output increases rapidly as the room cools and the temperature of the inlet air is decreased. A study of the curves in Figs. 2 to 8 and of Table 1 indicates that in general the convectors having thin heating units, or ones in which the vertical distance from the bottom to the top of the heating unit is small, were characterized by performance curves having the higher values of n, while the ones having the thicker heating units had lower values of n.

The effect of varying the height of the cabinet used with the same heating unit, for three types of heating units, is shown in Figs. 2, 3 and 4. As the height of cabinet was increased the total heat output was increased in each case. The values of n for the performance curves also changed. The relation between the value of n and the height of cabinet for the two types of convectors represented by Figs. 2 and 3 is shown in curves *A* and *B* in Fig. 9.

In each case the value of n attained a maximum with some intermediate height of cabinet. This height, for which the value of n becomes a maximum undoubtedly represents the optimum height of cabinet for the particular heating unit, since at this height the heat output increases the most rapidly per degree Farenheit reduction in the temperature of the inlet air. It may be observed from Fig. 9 that with the thinner or lower heating unit (curve *A*) the maximum was attained with a lower cabi-

net than it was with the higher heating unit (curve B). The two points for convectors Nos. 1 and 18 which had the highest heating unit also exhibit this same tendency, although no data on intermediate heights of cabinet were available.

The relative increase in the velocity of the air through the heating units would be determined by the height of the cabinet. The heat transfer would be influenced by the length of the path through the heating unit, or the height of the heating unit. While additional surface added to the height of the heating unit becomes less and less effective, it is probable that as the velocity of the air through the heating unit is increased a critical velocity will be reached above which the rate of heat transfer will decrease more rapidly than the relative rate of increase in velocity unless more heating surface is added. This may serve to explain why the maximum value of n was attained with lower cabinets in the case of the thin heating units than in the case of the thicker heating units. That is, the critical velocity would probably be lower in the former case than in the latter.

In Fig. 10 the heat outputs at the standard temperature difference of 150 F read from the curves in Figs. 2 and 3 are shown as curves A and B respectively, plotted against the corresponding stack heights. In both cases the observed points fall exactly on well defined smooth curves. This leads to the conclusion that if such a curve can be established with a limited amount of test data for any particular type of heating unit the equation for this curve can be used to obtain the rating of this heating unit with different heights of cabinets without the necessity for obtaining test data on all of the possible heights.

A general type of equation representing both curves shown in Fig. 10 is of the form

$$E = a + bh - c\,(z - h)^m \qquad (4)$$

where E = the heat output, Btu per hour
h = the stack height, inches
a, b, c, z, and m = constants determined by the particular type of convector

In the case of the type of convector shown in Fig. 2 and represented by curve A in Fig. 10, equation (4) becomes:

$$E = 5240 + 49.9h - 0.461\ (5 - 0.1h)^6$$

In the case of the type of convector shown in Fig. 3 and represented by curve B in Fig. 10, equation (4) becomes:

$$E = 4290 + 35.7h - 0.969\ (5 - 0.1h)^5$$

The effect of the length of the heating unit with the same height of cabinet is shown in Fig. 5. As the length was increased from 20 in. to 32 in. the value of n for the performance curve was decreased from 1.72 to 1.67. This decrease is small and it is evident from Table 1 that the length had very slight influence on the error resulting from the application of the correction factor.

The effect of the width of the heating unit is shown in Fig. 6. For the type of convector represented by Nos. 11 and 20 an increase in width of the heating unit increased the value of n while for the type represented by Nos. 6 and 23 an increase in the width of the heating unit decreased the value of n. Hence while the width apparently has some influence on the error in the application of the correction factor its effect is small and the nature of the influence is dependent on the type of convector.

In order to make the record of tests complete in this paper certain numbered points representing tests made in the low temperature testing plant or cold room have been shown in Figs. 2, 4, 7 and 8. The heat outputs for these tests may be compared with the heat outputs for the corresponding convectors in the warm wall booth by reading the latter from the curve at the same temperature difference. A complete discussion of these tests has been given in a previous paper and will therefore not be repeated. The conclusion drawn from the tests was that the heat output of a convector under service conditions in an actual room will be within 10 per cent of the heat output determined from tests in a warm wall booth, with the same temperature for the steam and for the inlet air in both cases, provided that the size of the convector is sufficient to heat the actual room to a temperature of approximately 68 F at the 30-in. level.

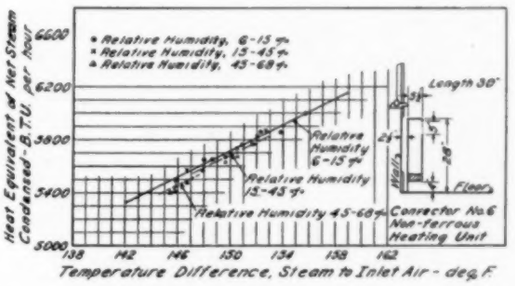

Fig. 11—Effect of varying relative humidities upon the heat loss from a convector with a non-ferrous heating unit

Results of Tests at Different Relative Humidities

The results of the tests run to determine the effect of the relative humidity of the surrounding air on the heat output of convector No. 6 are shown in Fig. 11. The base curve was determined by the points represented by the open circles and was established for relative humidities varying from 6 to 15 per cent. Three of the points indicated by crosses and representing relative humidities varying from 15 to 45 per cent were on the curve and five points were slightly below the curve. The curve established by these points was approximately 0.5 per cent below the base curve. One of the points indicated by triangles and representing relative humidities varying from 45 to 68 per cent was practically on the base curve and three points were below the curve. A curve established by these points was approximately 0.8 per cent below the base curve. Hence, while the results indicate a slight tendency for high relative humidities to decrease the heat output of a convector with the same temperature difference between the steam and the inlet air, it is evident that relative humidities as high as 68 per cent do not decrease the output to exceed one per cent.

With the standard frame construction used in the walls of the test room it was found impossible to maintain relative humidities higher than 68 per cent with temperatures in the cold room ranging from +2.0 to —6.0 F. Condensation started on the windows when the relative humidity rose to approximately 20 per cent, and began to appear on the inside surfaces of the exposed walls when the relative humidity reached 50 per cent. When the relative humidity exceeded 60 per cent

the inside surface of the exposed walls below the 7-ft level was thoroughly wet, water vapor was condensing on the baseboards, and water was running down the panels of the door. The glass in the windows was coated with approximately one inch of ice. The experience with this room indicates that it would be ruinous to attempt to maintain relative humidities higher than 40 per cent in buildings of frame construction when the outdoor temperature remains at zero, or below zero, for any considerable length of time.

The results of the tests run to determine the effect of

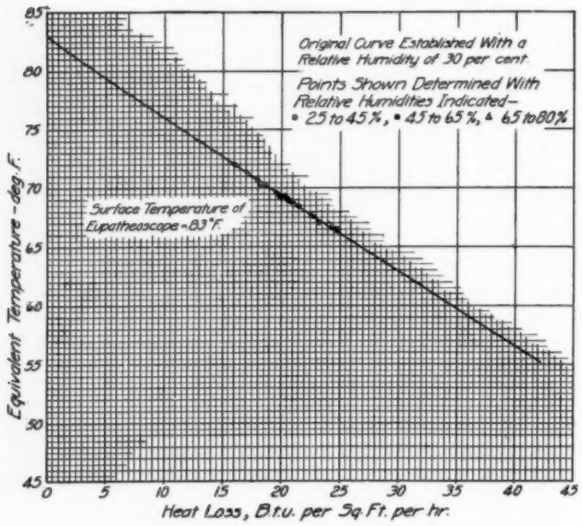

Fig. 12—Effect of varying relative humidities upon the heat loss from the eupatheoscope

the relative humidity of the surrounding air on the heat loss from the eupatheoscope are shown in Fig. 12. From this it is evident that all of the points representing relative humidities from 25 to 80 per cent are on the same calibration curve for the eupatheoscope, indicating that relative humidity has no measurable effect on the heat loss from the instrument. That is, in a given uniform environment with walls and air at the same temperature, no appreciable change in the heat loss from the eupatheoscope occurs with variations in the relative humidity of the air. However, relative humidities of 80 per cent or more affected the insulation of the electrical circuits, and resulted in making doubtful the reliability of the temperatures as observed by means of the thermocouples.

Conclusions

The following conclusions may be drawn from the test results:

(1) The relation between the heat output and the difference in temperature between the steam and the inlet air for convectors may be expressed by a general equation of the form:

$$H = K\ (t_s - t_1)^n$$

(2) A correction factor $C = \left[\frac{150}{t_s - t_1}\right]^{1.5}$, for reducing the heat output of a convector under test conditions to the heat output under standard conditions, with steam temperature of 215 F and inlet air temperature of 65 F, is applicable within a probable error of 3 per cent for temperatures of air at inlet between 60 F and 75 F.

(3) The error resulting from the use of the exponent 1.5 in the correction factor is less than that resulting from the use of an exponent of 1.3 as now recommended in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[3].

(4) The value of n for the performance curves and the magnitude of the error in the application of the correction factor is influenced by the type and physical dimensions of the convector. The exact nature of the error is uncertain, and it is difficult to predict whether it will increase or decrease when the dimensions of the convector increase or decrease.

(5) There is a slight tendency for the heat output of a convector with the same temperature difference between the steam and the inlet to decrease as the relative humidity of the air is increased; but relative humidities as high as 68 per cent do not result in a decrease to exceed one per cent of the heat output obtained with relative humidities of from 6 to 15 per cent.

(6) In a given uniform environment with walls and air at the same temperature, no appreciable change in the heat loss from the eupatheoscope occurs when the relative humidity of the air varies over a range between 25 per cent and 80 per cent.

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[2]A. S. H. V. E. Research Paper entitled Tests of Convectors in a Warm Wall Testing Booth, Part II, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick, A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August, 1933, p. 431.

[3]A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code), A. S. H. V. E. TRANSACTIONS, VOL. 37, 1931, p. 367.

[4]A. S. H. V. E. Research Paper entitled The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz and M. K. Fahnestock. A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1933, p. 369.

[5]University of Illinois Engineering Experiment Station *Bulletins* Nos. 192 and 223.

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[7]A. S. H. V. E. Research Paper entitled Performance of Convector Heaters, by A. P. Kratz and M. K. Fahnestock. A. S. H. V. E. TRANSACTIONS, VOL. 38, 1932, p. 351.

[8]Surface Heat Transmission, by R. H. Heilman (Trans. *A.S.M.E.*, Fuels and Steam Power Section, Vol. 51, No. 22, September-December, 1921, pp. 287-302).

[9]The Basic Laws and Data of Heat Transmission, III Free Convection, by W. J. King (*Mech. Eng.*, Vol. 54, No. 5, May, 1932, pp. 351-354).

A. S. A. Standard for Lantern Slide Charts

A standard has recently been approved by the *American Standards Association* in the form of an American Recommended Practice for Engineering and Scientific Charts for Lantern Slides (Z15.1-1932).

The standard, which provides rules for widths of lines, lettering, and symbols designed to assure a maximum of legibility, was developed by a special sub-group of subcommittee 4 on Engineering and Scientific Graphs, of the Sectional Committee on Standards for Graphic Presentation (Z15), sponsored by the *American Society of Mechanical Engineers*, and is available from the *American Standards Association*, 29 West 39th St., New York, N. Y.

Dry Bulb vs. Effective Temperature Control

By **A. E. Beals, Norwich, N. Y.**

(NON-MEMBER)

THE determination of effective temperatures and the location of human comfort zones[1] upon the psychrometric chart have been recognized as notable scientific contributions and the Society is to be commended for sponsoring the long series of studies which have aided so materially in the rapid development of air conditioning for comfort. Well-merited appreciation is also due to those workers, whose skill and ingenuity has brought to such an orderly conclusion the mass of data and figures, which necessarily accrued during the years of investigation. All of the facts, figures and conclusions have been generously made public by the Society for unrestricted use by those interested in the progress of air conditioning.

In an early report we find an ingenious chart, devised to show at a glance the relations between dry- and wet-bulb temperatures and the resulting effective temperatures; also that experiment has clearly demonstrated that dry-bulb temperatures, per se, are wholly meaningless in terms of bodily sensations to heat and cold.

Of what practical use is all of this information and knowledge which has been collected and recorded so painstakingly? What are engineers doing with it?

A study of the comfort data shows that an effective temperature of 71 deg, within certain limits of relative humidity, is the summertime condition of maximum comfort[2] for 98 per cent of the subjects tested. Further, it was found that only about 50 per cent of the subjects felt comfortable at effective temperatures of 67½ deg on the cool side, and 75 deg on the warm side. It should be noted also that the subject must remain in and become inured to this environment. This acclimatization requires from two to three hours, depending upon the previous surroundings. Meanwhile a considerable degree of discomfort may be experienced from the contrast, particularly when the change is made from a higher to a lower level of temperature. Even when time is allowed for acclimatization, the range of comfort for the majority of people is exceedingly narrow, when measured in terms of effective temperatures. Likewise, the permissible drop in effective temperature conditions is also exceedingly narrow if undue shock is to be avoided when changing from one to the other. In passing from a condition of high temperature or high humidity, and perhaps both, into a cooled room, an additional drop of only 1 deg of effective temperature will make an appreciable difference in the contrast and comfort experienced.

For instance, from a condition of 82 deg effective temperature, one will experience a much greater degree of temporary comfort, at an effective temperature of 76½ deg than at 74 deg if the relative humidity is at the proper point. In fact, an effective temperature of 76½ deg by contrast, would seem quite comfortable, if the relative humidity was such that perspiration was not unduly evidenced. Therefore, homes, expecting visitors, or offices, desiring business callers, should not be constantly maintained under a condition of maximum comfort, regardless of prevailing outside circumstances. This fact was rather belatedly recognized in the case of theaters, auditoriums and department stores, and a code[3] was devised governing the cooling of such places.

The A. S. H. V. E. Ventilation Standards stipulate that effective temperatures between 64 and 69 deg shall be maintained during the winter season, and between 69 and 73 deg during the summer when cooling is required. They also provide that the relative humidity shall not be less than 30 per cent nor greater than 60 per cent.

Considering only the summer conditions, this code covers quite a large area upon a psychrometric chart but the range of effective temperatures is small. As the conditions cited apply only to cases where the bodily sensation has reached equilibrium with the surrounding air, no consideration was given to outside temperatures and humidities, nor to the possible contrast in passing from one to the other. The need, however, for reducing the contrast between the outside and inside conditions was evident. Therefore, as a concession, it was further recommended that the inside dry-bulb temperature be maintained at 72 deg, plus one-third of the difference between the outside dry-bulb temperature and 70 deg.

In the light of all the information gained during the research into effective temperatures, it is a perfectly natural question to ask, why the outside dry-bulb temperature alone should, in any way, have a bearing upon the inside dry-bulb temperature. And a still more pertinent question would be, why should the outside *dry-bulb* temperature alone govern the inside *effective* temperatures.

If the dry-bulb temperature of itself is not an indication of bodily sensation on the inside of a theater, neither is it any criterion of the sensation experienced on the outside. Therefore, the relations of the dry-bulb temperatures, within and without an auditorium, in no way measure the contrast experienced by the human body in passing from one to the other.

Nevertheless, this code completely ignores the adaptability and use of the principles of effective temperatures. It recommends that the control for inside conditions be based entirely upon the relation between the dry-bulb temperatures. It merely stipulates that the inside effective temperatures be maintained within certain limits.

[1]A. S. H. V. E. Transactions, Vol. 29, 1923, p. 163, p. 361.

[2]A. S. H. V. E. Transactions, Vol. 35, 1929, p. 269.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Buck Hill Falls, Pa., June, 1934.

[3,4]A. S. H. V. E. Transactions, Vol. 38, p. 383.

Still loyal to the old dry-bulb tradition, a table[4] based upon dry-bulb temperatures, giving the effective temperatures at different percentages of relative humidity, was published. Is that table of any particular benefit to the engineer, trying faithfully and conscientiously to operate his cooling plant to best advantage? If any attention whatever is to be paid to effective temperatures, this table might better have been based upon such temperatures, giving both dry- and wet-bulb values for the recommended inside effective temperatures and percentages of relative humidity. The engineer then would have a chance to at least check up and determine whether or not he was still within bounds.

To show the indefiniteness of this table, it is only necessary to examine the range of effective temperatures allotted to the dry-bulb temperature of 77 deg. These effective temperatures comprise practically the entire range from the lower to the upper allowable limits, the wet-bulb temperatures ranging from 57½ to 67 deg.

Also, according to the rule for auditoriums and the like, the inside temperature of 77 deg is that required for an outside temperature of 85 deg. But an outside dry-bulb temperature of 85 deg might be attended by any wet-bulb temperature between 64 and 80 deg, with a corresponding range of effective temperatures from 75 to 82 deg.

Imagine the contrast in going from an outside effective temperature of 82 deg into one of 69.7 deg! Furthermore, it is certain that one would feel much more comfortable outside at a temperature of 85 deg dry-bulb and 75 deg effective temperature than on the inside at 77 deg and an effective temperature of 72.6 deg.

Coincident with the adoption of the code by the Society a committee published an explanation of the use and application of the effective temperature chart.[5] This committee went even further than the code in repudiating the principle of effective temperatures. In its recommendations of conditions to be maintained indoors, for exposures of less than three hours, the use of effective temperatures was discarded altogether—inside as well as outside. It is the committee's suggestion that the inside dry-bulb temperatures vary in accordance with the same "rule of thumb" formula as expounded by the code, with the additional limitation that a constant inside dew-point of 57 deg be maintained. Thus, after a clear and concise exposition of the effective temperature chart and comfort zone, this committee casts aside its practical application in air-conditioning, and relegates this really monumental piece of research work to the status of an academic curiosity.

The constant dew-point of 57 deg for inside conditions was evidently chosen because, within the recommended range, a line, indicating the various conditions at this dew-point, lies well within the area of the comfort zone as defined by the code. But the resulting effective temperatures along this line are more in accord with permanent comfort, to which one must become acclimated, than with temporary comfort in contrast to the prevailing outside conditions.

However, the maintenance of inside conditions at any constant dew-point is merely a pleasing fiction. In actual practice, with the usual methods of control now employed, this "just isn't done."

[5-6] A. S. H. V. E. Transactions, Vol. 38, 1932, p. 410.

A thermostat, placed near the outlet of a washer, and set to maintain a constant wet-bulb temperature of the air leaving it, is incapable of maintaining a constant dew-point in an auditorium, subject to varying ratios of moisture to heat to be removed. For instance, assume that the dry-bulb thermostat is set to maintain a temperature of 72 deg in the room, and it is attempted to hold a constant dew-point of 57 deg. Also assume that a ratio of 1.37 grains of moisture per 1 Btu of heat has to be removed. To accomplish this, the saturated temperature of air leaving the washer would need to be 54 deg.

Suppose now that the load changes to such an extent that the ratio of moisture to heat removed becomes 0.75 grains per Btu. In order to maintain the 72 deg dry-bulb temperature at 57 deg dew-point in the room, it would then be necessary for the engineer to reset his thermostat for 55½ deg. If the outside temperature changes to 85 deg and the conscientious engineer consults his table, he finds that the inside temperature should be 76½ deg with 64 deg wet-bulb. Accordingly he immediately (?) resets his dry-bulb thermostat. Then he wonders what's going to happen.

As the inside temperature goes up, the ratio of moisture to heat removed also changes until it reaches 1.74 grains per Btu. To meet this demand, the wet-bulb thermostat again requires resetting for a saturated air temperature of 50¾ deg. But how is the engineer to know that?

By the time the harassed engineer thinks he is all set, maybe his inside load has changed again so that it is necessary to remove only one grain of moisture per Btu of heat. This requires a saturated air temperature leaving the washer of 54½ deg. And again, how is the engineer to know that?

Changes of the outside conditions and variations of the inside requirements are constantly taking place. The audiences in a theater are never there long enough to permit their bodily sensations to come into equilibrium with the environment. Under the rules and regulations promulgated by the code and the Society's committee, it is by accident, pure and simple, if a theater is ever found to be, for even a few fleeting moments, in a condition approximating temporary comfort. Many times, when one leaves, he wishes he had not gone in, and oftentimes, when one enters, he is tempted to back out, but to save the price of admission he decides to suffer and possibly contract a severe cold.

In actual operation of an air conditioning plant, instead of a constant dew-point of 57 deg within a theater, a constant wet-bulb temperature of the air leaving the washer is maintained. If the inside dry-bulb temperatures are varied with the outside dry-bulb temperatures, the ratio of moisture to heat required to be removed determines the dew-point, which varies accordingly. At the same time the wet-bulb and the effective temperatures vary also.

Another table, supplementing that of the code and setting forth "desirable indoor air conditions," has been published.[6] This table is based upon an inside, constant dew-point of 57 deg, and the inside dry-bulb temperatures are varied with the outside dry-bulb temperatures. The relations between these three temperatures have no bearing upon comfort sensations, and the effective temperature relations are uncontrolled.

In practice no attempt is made to maintain a constant,

inside dew-point. In cases where the dry-bulb temperature recommendations are attempted, the wet-bulb and effective temperatures are entirely out of control of the cooling apparatus. If, however, a constant wet-bulb temperature of air leaving the washer is maintained at 50 deg, the comfort conditions established by this table will, under inside, full load requirements, be fairly well approximated. Under half-load requirements, the comfort conditions are not so good. The inside dew-points will vary from 51 to 60 deg for outside temperatures ranging from 90 to 100 deg; for outside temperatures from 74 to 80 deg, the dew-points will vary from about 52½ to 54½ deg. In the final analysis of its practical application to air conditioning, it is thus evident that this table merely recommends that inside *dry-bulb* temperatures be maintained in accordance with varying outside *dry-bulb* temperatures—and nothing more.

It is true that with a properly chosen wet-bulb temperature of air leaving the washer, the inside conditions can be held well within the established, average comfort zone. It is, however, a debatable question whether the conditions established by varying the dry-bulb temperatures are best either for the temporary comfort of the patrons or the financial interest of the theater owner. The costs for power, water and attendance, in the operation of this class of equipment, is one of the large factors in deciding whether the apparatus shall be installed or not. The temporary comfort of the patrons is a large factor in determining the amount of box office receipts. Therefore, the interests of all concerned will best be served when the maximum temporary comfort of the patrons can be assured at a reasonable cost of operation.

In public buildings, the criterion for inside temperature and humidity is the contrast experienced in passing from one condition to the other. This contrast is measured solely by the relation which the inside *effective* temperature bears to the outside *effective* temperature. The contrasting *dry-bulb* temperatures, taken by themselves, have no measuring nor comparative values whatever.

The wet-bulb temperature, prevailing on the outside, is a factor of just as much importance as the wet-bulb temperature, assumed to be maintained inside, or as either of the dry-bulb temperatures. As for the inside dew-point, that means nothing whatever.

Other influences, besides the actual temperature and humidity of the outside air, have an important bearing upon the degree of contrast experienced upon passing into a cooled room. Before entering a theater, the patrons have been exposed to the radiation from hot pavements and building walls, and the majority have indulged in more or less bodily activity. Each accentuates the sensation of warmth. After becoming seated, a period of relaxation ensues and all bodily activity practically ceases—unless involuntary shivering is produced due to too rapid withdrawal of heat. Therefore, for temporary comfort, too great a difference between the outside and inside effective temperatures should not be permitted. Neither should the relative humidity be so low that too rapid evaporation of perspiration is induced.

A number of years ago the writer saw an architect's specifications for a large auditorium which stipulated that the capacity of the air conditioning equipment should be sufficient to maintain, at all times, "an inside effective temperature midway between the outside effective temperature and an effective temperature of 66 deg." This, so far as the writer is informed, was the first and only recognition of the practical adaptability of the principle of effective temperatures. This proposal was made and the limiting temperature of 66 deg was chosen before later investigations had determined that 71 deg effective temperature is the condition of maximum summer comfort.

In the light of later investigations, an effective temperature midway between the outside effective temperature and an effective temperature of 71 deg would appear to be a reasonable compromise between the condition of maximum comfort and the discomfort of the outside environment. It would insure relief from the outside conditions and a sense of temporary comfort without undue shock, when passing either in or out.

The relative humidity should follow rather closely the "perspiration line."[7] This, through the required range, approximates 60 per cent for the lower and 50 per cent for the higher temperatures. Thus, while perspiration would not be unpleasantly evident, the rate of evaporation would not be sufficient to foster a sensation of chilliness. A regulation of this nature, which is based upon scientific principles governing inside conditions to be maintained, would seem to be ideal for short time exposures. It is also a regulation which is susceptible of 100 per cent automatic control.

With complete automatic control the human equation is entirely eliminated from the operating problem, and with the inside effective temperatures continually varying in accordance with the outside effective temperatures, a pleasing sensation of change will unconsciously be experienced. Also, regardless of sudden and wide variations of outside conditions, the inside environment will always be commensurate therewith, and extreme contrasts, in passing either in or out, will be avoided. The old time slogan of "Twenty Degrees Cooler Inside" has been entirely discredited.

Thus far only the comfort of patrons has been made the paramount consideration, but the trouble and expense to be borne by the owner in the rather complicated operation of air conditioning equipment is also entitled to consideration.

It is an easily ascertained fact that there is not a refrigerating unit used at present in connection with comfort cooling that is operating to good advantage. This is not the fault of the refrigerating equipment, which is usually of the best, nor is it the fault of the refrigerating engineers and designers. It is because of the requirements imposed by the air conditioning engineers.

One factor contributing to this state of affairs is the general practice of maintaining a constant wet-bulb temperature of the air leaving the washer. Another is the idea that it is better to intensively dehumidify a portion of the recirculated air. They are both at fault from the viewpoint of obtaining refrigerating economy. The first entails the maintenance of a constant temperature at which the refrigerating equipment must absorb heat. The second necessitates that this temperature be kept at the lowest point required by the maximum ratio of moisture to Btu of heat to be removed.

A refrigerating machine is an apparatus which absorbs

[7] Where Is the Perspiration Line? A. E. Beals, *Heating, Piping and Air Conditioning*, Apr., 1933.

heat at a lower temperature level and discharges it at a higher level. The high level of discharge is fixed within a relatively small range by the temperature of the available condensing water supply. Therefore, the lower the temperature at which the heat must be absorbed, the

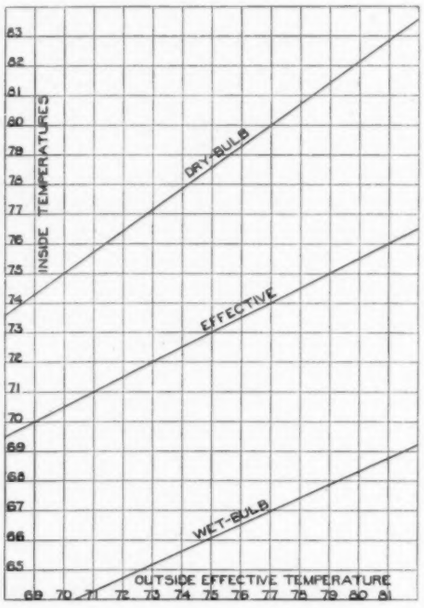

Fig. 1—Suggested inside conditions to be maintained corresponding to various outside effective temperatures

greater will be the power required. There are other factors which also influence the economical operation of the refrigerating unit, not necessary to discuss here.

By automatically controlling the inside effective temperatures in accordance with the outside effective temperatures, the maximum of temporary comfort can be provided for the patrons. By utilizing this same control to vary, in accordance with the requirements, the temperature at which the refrigerating machine must absorb its heat, a very considerable saving in costs of power and water can be effected. The average saving, over the present methods of operating, will at a conservative estimate amount to at least 30 per cent. As this saving is predicated upon maintaining a greater degree of temporary comfort, the best interests of both the patrons and the owner are thus assured.

The chart, Fig. 1, shows suggested inside conditions to be maintained, corresponding to various outside effec-

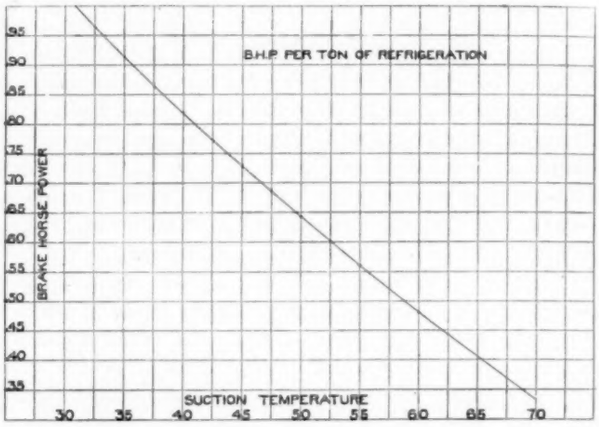

Fig. 2—Brake horsepower per ton of refrigeration when unit is operating under various suction temperatures

tive temperatures. This chart is based upon an inside effective temperature, which shall be midway between the momentary outside effective temperature and an effective temperature of 71 deg. The dry- and wet-bulb temperatures are so arranged that at an inside effective temperature of 71 deg, the relative humidity will be 55 per cent; at an inside effective temperature of 76 deg, the relative humidity will be 50 per cent.

As an indication of one source of saving in cost of operation, Fig. 2 is presented for consideration. This chart shows the approximate brake horse power per ton when the refrigerating unit is operating under various suction temperatures. The values shown are conservative because a 15 per cent heat loss from the refrigerating apparatus has been assumed, and the efficiency of the compressor has been figured at 85 per cent.

It is evident from Fig. 2 that the higher the temperature at which it is possible to utilize the cooling water in an air washer, the less will be the power cost.

Classification of Coals

A new standard on the classification of coals, published by the *American Standards Association,* specifies how much heat the consumer may expect from a dollar's worth of any kind of coal. This work was begun seven years ago by a group of operators, industrial consumers of coal, and Canadian, federal and state government experts, under the auspices of the *A.S.A.,* the technical work being directed by the *American Society for Testing Materials.*

The committee subdivided the four broad types of coal, i.e., anthracite, bituminous, subbituminous, and lignite, into 13 groups. The minimum amount of heat per ton, the maximum amount of impurities per ton and other standards for each of these groups of coal were established as requirements.

Through the entire project the committee kept in mind the necessity for developing usable requirements or standards, practical for mining interests, distributors, and consumers. Large industrial buyers of coal, such as power plants, railroads and factories, and the domestic consumer will now have a guide in buying fuel.

These specifications are being sent to NRA Coal Code authorities in every section of the country for use as the basis for their work in classifying the output of mines.

A. C. Fieldner, U. S. Bureau of Mines, served as chairman of the committee, H. J. Rose, Mellon Institute, was vice-chairman, and C. B. Huntress, *National Coal Association,* was secretary.

Copies of the complete specifications, known as the Classification of Coals by Rank and by Grade, may be obtained from the *American Society for Testing Materials,* 260 S. Broad St., Philadelphia, Pa., or from the *American Standards Association,* 29 West 39th St., New York, N. Y.

Criteria for Industrial Exhaust Systems

By **J. J. Bloomfield*** ***(NON-MEMBER)***

Washington, D. C.

KNOWLEDGE of the requirements of industrial exhaust systems, for the effective removal from the air of toxic dusts, fumes, gases and vapors, in an effort to maintain hygienic conditions in workrooms, is of importance to the engineer. Although the installation and use of correctly designed exhaust systems are essential in the maintenance of hygienically safe conditions, they are not in themselves a complete solution of the entire problem. Exhaust ventilation devices, like any other machinery, require constant care if adequate protection is to be afforded to the industrial worker. Hence, it is essential that such systems be tested from time to time, in an effort to determine how effectively they are functioning and if they are being maintained properly. Such an evaluation requires the actual determination of those substances in the air which the system was designed to eliminate. The present paper treats in a general manner some of the methods and instruments which may be used in testing the efficiency of industrial exhaust systems.

The classification of dusts, fumes and smoke, on the basis of their physical properties given in Chapter 15 of the A. S. H. V. E. GUIDE 1934 will not be amplified in this discussion except that dusts will be discussed from the viewpoint of their action on the body; namely, those which upon inhalation come in contact with the blood stream and exert a generalized toxic effect (compounds of lead, arsenic, cadmium, etc.), and dusts which produce a localized reaction, such as certain fibrosis-producing dusts exemplified by silica, asbestos, etc., which usually confine their deleterious action to the lungs.

The Sampling and Analysis of Dusts

Fibrosis-producing dusts: The abundant evidence at hand showing that the inhalation of certain industrial dusts is an important factor in the causation of pulmonary disease has emphasized the significance of the quantitative aspects of this problem. A knowledge of dust content of the industrial atmosphere is required not only for the purpose of determining the extent of the hazard involved in various manufacturing processes, but is also useful in measuring the efficiency of protective devices which may be used in the elimination of the dust hazard.

The properties of a given dust which determine its capacity to produce pulmonary pathology are the nature of the dust, that is, its chemical and mineralogical composition, its particle size and finally the quantity of the dust dispersed in the atmosphere.

One of the outstanding results of the last 20 years of research in the field of dust inhalation is the demonstration of the fact that, in general, the degree of health hazard associated with the inhalation of any dust, all other factors remaining constant, is dependent upon the mineralogical composition of the dust. For example, it is now established that the inhalation of certain types of dust, such as granite dust (1), will in time produce fibrosis of the lungs, frequently associated with tuberculosis. In other cases exposure to dust may result in the production of a far lesser degree of fibrosis without subsequent tuberculosis; this is true of cement dust (2). And finally, there are certain types of dusts which produce little lung fibrosis, as typified by marble dust (3). In general it has been found that those dusts which are high in quartz content are the ones which most readily produce a disabling fibrosis of the lungs. Hence, the necessity for knowledge concerning the chemical and mineralogical composition of a dust is obvious.

So far as the size of the dust particles is concerned, it is apparent that in order for any given dust to produce injury to the lung it must gain access to the parenchyma of the lung, the site where the harmful effects of the dust take place. It is known that not all of the particles of inhaled dust gain access or are retained by the human lung (4). For this reason it is of value to determine the size of the dust particles present in the industrial atmosphere.

With reference to the quantity of dust present in the air of a workroom it is apparent that when the dust concentration is high the exposed person will inhale a greater quantity in a given period of time than he will when the dust concentration of the atmosphere is relatively low, and since the rate of production of the disease is partially dependent upon the total amount of dust inhaled, this latter fact plays an important role in determining the time of onset of the end result. The need for the evaluation of the quantity of dust in the industrial atmosphere is obvious.

Composition of industrial dusts: Present day knowledge of the effects on the lungs of inhaled dust is more complete for quartz-containing dusts than for any other type of particulate matter. In general, it has been found that the harmfulness of a quartz-containing dust is in direct proportion to its quartz content. For this reason in attempting to evaluate the harmfulness of a dust it is of the utmost importance to ascertain its quartz content. This is best done by a combined chemical and petrographic analysis (5). It has been found in practice that samples of dust settled out of the atmosphere at the breathing level of the worker serve admirably for mineralogical analyses, which are best carried out by a competent geologist. Since no two dusts offer the same problem it is difficult to specify general procedure for such an analysis. Suffice it to say that each sample must first undergo a careful examination under the petrographic microscope, and in addition is further subjected to a complete chemical analysis with frequent petrographic examinations throughout the entire process. Only by such an analysis has it been found possible to determine accurately the percentage of quartz present in quartz-containing dusts. Table 1 presents the quartz

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content of dusts obtained in various industries in which the industrial dust problem has received some study.

Size of Dusts: It has been demonstrated that particles of a size greater than 10 to 12 microns in longest dimension are very seldom found in the lungs. This absence

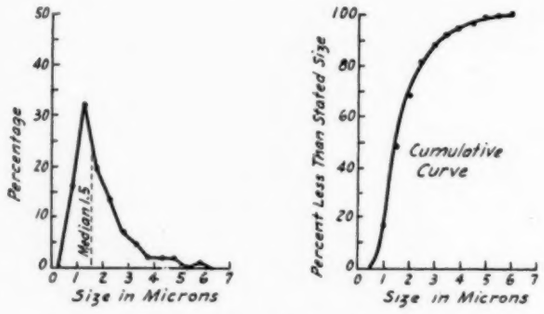

Fig. 1—Particle size distribution of talc dust

of larger particles is partly due to the fact that the numbers of such particles greater than 10 microns in size present in industrial air is, as compared with the lower sizes, comparatively small and due to gravity and the protective action of the mucous surfaces of the upper respiratory tract, these larger particles do not penetrate to the terminal portions of the respiratory tract. Hence attention need only be given to those dust particles which are less than 10 microns in longest dimension.

Table 1—Percentage of quartz present in various industrial dusts

	Percentage of Quartz
Rock drilling dust (bituminous coal mine)	54.0
Granite cutting dust	35.2
Rock drilling dust (anthracite coal mine)	31.0
Brass foundry dust	19.0
Dust from raw mills in cement plant	6.5
Slate mill dust (Vermont red slate)	3.0
Silverware polishing dust	1.7
Anthracite coal dust	1.5
Bituminous coal dust	1.2
Cement dust	1.0
Slate mill dust (Vermont green slate)	trace
Talc mill dust	none
Marble cutting dust	none

In order to ascertain the size distribution of an industrial dust it is necessary to make particle-size studies of the dust under consideration. In practice the samples for such studies may be obtained by the use of the Owens Jet Dust Counter (6). This apparatus projects the atmospheric dust directly on a microscope cover-slip. This cover-slip may then be properly mounted and examined microscopically. The particle size dimensions which are obtained from the microscopic study, are grouped in classes according to size, from which a percentage distribution curve is easily obtained. Fig. 1 presents the particle-size distribution of talc dust from the air of a workroom in which talc was being ground to a very fine state of subdivision. The measurements, from which the data for this figure were obtained, were made by means of an occular filar micrometer at a magnification of 1,000 diameters (7). With this magnification it is possible to measure particles as small as 0.5 microns in diameter, while particles smaller than 0.5 microns are easily distinguished at this magnification and although not measured their presence is recorded.

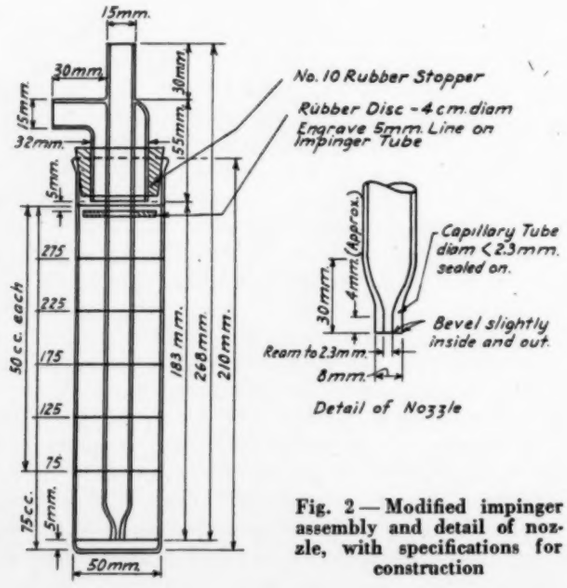

Fig. 2 — Modified impinger assembly and detail of nozzle, with specifications for construction

Quantity of Dust: As pointed out earlier a knowledge of the quantity of dust dispersed in the industrial atmosphere is very important, since with any given dust the rate of production of the injury will be partially dependent upon the total quantity of dust inhaled.

Many methods have been devised and used for the purpose of determining the quantity of dust in air (8) (9). For the purpose of dust sampling in either high or low dust concentrations, the Greenburg-Smith Impinger apparatus now finds universal favor (10). This instrument has been used by the United States Public Health Service in all of its dust studies during the past eleven years and is also being used by other workers in this country and abroad.

In this instrument, the air to be sampled is drawn through a glass tube and impinged at a high velocity on the flat bottom of a glass flask containing water or other suitable fluid. The dust is momentarily arrested, wetted by the collecting fluid, and in this manner trapped.

The impinger apparatus consists essentially of two portions; *first,* a source of sufficient suction to draw the air to be sampled through the sampling device; and *second,* the sampling device or impinger itself, which consists of a container and the impinger tube. As a source of suction one may use either an electrically-driven pump or a compressed air ejector device (10) (11). Fig. 2 depicts the essential portions of the impinger device, which consists of a straight piece of Pyrex glass tubing 15 mm in outside diameter and approximately 275 mm in length. The tube is drawn down in stream line form at its lower end, to a tip with a 2.3 mm orifice. In sampling this orifice is kept at distance of 5 mm from the bottom of the flask. The suction connection is combined with the inlet tube. The flask is 50 mm in diameter and 210 mm in height and requires a fluid (water) volume of only 75 cc to give the proper depth of immersion to the nozzle. An entrainment trap in the form of a rubber ring prevents the possible loss of the liquid (and dust) with the outgoing air.

In sampling, the outlet or suction elbow of the sampling flask is connected with the source of suction by means of a suitable length (25 ft) of non-collapsible rubber tubing. The duration of the sampling period should be such as to yield a satisfactory suspension of dust for analysis and is thus dependent on the concentra-

tion of dust in the atmosphere. Under the usual industrial conditions, samples of from 10 to 30 cu ft of air yield sufficient suspended dust for analysis. Since a sampling rate of 1 cu ft per minute is maintained, this will require a sampling period of from 10 to 30 min.

The collecting efficiency of the apparatus is dependent upon adherence to the previously cited impinger tube dimensions and the sampling rate of 1 cu ft of air per minute. Experimental tests of this instrument against finely-divided silica dust suspensions in air have consistently yielded efficiencies of 98 per cent at the specified sampling rate (11, 12).

Since practically all dusts are, to some extent, soluble in water, it is good practice to analyze the samples as soon as possible. Such practice tends to prevent any undue flocculation as well as any solvent action on the dust particles. In the laboratory the dust suspension in the sampling fluid may be filtered through a 325-mesh screen and then diluted so that the number of dust particles in the microscope field is equal to approximately

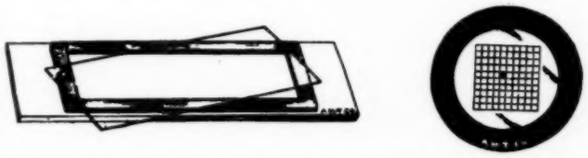

Fig. 3—(Left) Sedgwick-Rafter Cell. (Right) Whipple Disc

50 to 75. Two or more 1-cc portions are placed in Sedgwick-Rafter cells for counting (Fig. 3). The microscope is of the ordinary type provided with a suitable eyepiece and objective and fitted with an Abbé condenser. A Whipple disc eyepiece micrometer (Fig. 3) is placed in the microscope eyepiece and the microscope tube length is adjusted so that the side of the ruling in the eyepiece is 1 mm in length. (A 7.5 X eyepiece, 16 millimeters objective and a tube length of 178 millimeters has been found to give this result.) An ordinary type of microscope lamp may be employed as a source of illumination. In order to provide a high degree of visibility for refractile objects it is best to lower the Abbé condenser system below the usual focusing point and to restrict the opening in the iris diaphragm. In making counts the microscope should be focused throughout the depth of the cell since some of the dust particles may remain in suspension. Since the counting cell is 1 mm deep and the area in the microscopic field is 1 square millimeter each count represents the amount of dust in a cubic millimeter of the sampling fluid. Knowing the original dilution of the sample and the number of cubic feet of air sampled it is an easy matter to compute the number of dust particles in the sample per cubic foot of air. It is of course necessary to make control dust counts on the sampling fluid.

Table 2 presents a summary of the average dust content of the air in certain dusty industries.

Poisonous dusts. Some of the common poisonous dusts which exist in industrial atmospheres are the compounds of lead, cadmium, arsenic, mercury, etc. For the collection of these dusts one may employ any one of several instruments. For the collection of radioactive dusts (13), the paper thimble has been successfully used.

Table 2—Average dust count in certain dusty trades

Industry	Dust count in millions of particles per cubic foot of air
SLATE FINISHING MILLS	
Floormen	1598.0
Loaders	1276.0
Disc crusher operators	312.8
TALC MINING	
Jack-hammer drillers	2159.8
Muckers	44.8
TALC FINISHING MILLS	
Crushers and cylindermen	14.0
Packers	50.1
MARBLE CARVERS	19.1
MARBLE CUTTERS	32.8
GRANITE QUARRYING	
Leyner drillers	144.4
Jack-hammer drillers	112.1
Plug drillers	36.9
CEMENT MILL, average of all operations	26.0
GRANITE CUTTING	
Hand pneumatic tool operatives	59.2
Machine pneumatic tool operatives	35.9
Attendant labor	17.0
ANTHRACITE COAL MINING	
Miners and miners' helpers	231.5
Attendant labor	31.1
BITUMINOUS COAL MINING	
Coal cutters and coal loaders	112.3
Attendant labor	8.9
SILVERWARE MANUFACTURING	
Dusty processes	5.2
Non-dusty processes	1.7
MUNICIPAL DUST (STREET CLEANERS)	
Congested district	4.1
Residential district	1.8
COTTON INDUSTRY	
Carding room	8.6
Weaving and spinning room	4.6

In practice, a single thickness, Whatman extraction paper thimble, containing cotton wool, well fluffed out, is employed. Air is passed through such a thimble at the rate of 1 to 2 cu ft per minute, until sufficient dust has been collected, after which the contents of the thimble are subjected to a chemical analysis.

For precipitating dry, poorly conducting dust suspensions, non-rectified alternating current has been employed successfully (14). In such a device air is passed through a glass precipitating tube at a fixed rate, using 15,000 volts, the dust being generally collected on a celluloid foil inserted in the tube. The material deposited on the foil may then be examined by chemical methods.

For most purposes, however, the impinger apparatus, already described in detail, is well adapted for the sampling of suspensions of poisonous dusts commonly encountered in industry. The collecting medium may be water, or some other suitable fluid. Blank tests on the sampling medium should be made.

In determining the extent of air pollution by poisonous dusts, one needs to consider two factors, irrespective of the sampling instrument employed. These are the need of a sensitive and reliable chemical method for the estimation of the quantity of the polluting material, and a sufficiently large sample of air-borne dust must be collected to insure adequate amounts for the specific method of analysis employed.

The Sampling and Analysis of Fumes, Gases and Vapors

Fumes. The most common fumes encountered in industry are those originating from the oxidation of vapors formed by the heating of metals to high temperatures. Some of these fumes present in the air of workrooms are lead oxides, cadmium oxides, zinc oxides, etc. These fumes may be collected either by the electrical precipitating method (14) or by the impinger apparatus (10). In case the latter instrument is used it is advisable that two impinger containers be employed

in series. This precaution is necessary, since it has been found that the impinger device has a lower sampling efficiency for very finely divided materials, such as fumes, than for dusts (11). Once the sample has been collected it may then be subjected to chemical analysis. The impinger apparatus has been used for the collection of chromic acid mist, the collecting medium employed being a normal solution of sodium hydroxide (15).

Gases and Vapors. The gas and vapor hazards of modern industry are extremely important and the fatalities from poisoning by such substances increase year by year. Gases and vapors occur as raw materials, as products and as agents in manufacturing processes. Among some of the common gases one may list carbon monoxide, sulphur dioxide, hydrogen sulphide, hydrogen cyanide, ammonia, and some of the gases now used for refrigerating purposes, such as methyl and ethyl chloride. Many chemicals which are volatile at ordinary room temperatures, are used very extensively in industry in the form of solvents and lacquers. They constitute an extremely important group of contaminants and are characterized by their low density and easy volatility. Among some of the common vapors of a toxic nature found in industry are benzol, carbon disulphide, methyl alcohol, carbon tetrachloride and trichlorethylene.

Space does not permit the consideration of all of these gases and vapors. A discussion of one gas (CO), and one vapor (benzol) should suffice to illustrate the technic employed in determining the concentration of such substances in air.

There are several methods now available for the detection of carbon monoxide in air. Samples may be collected in glass containers by water displacement or by vacuum, after which they may be analyzed by either the Iodine Pentoxide method (16) or by the Pyrotannic acid method (17). In the former method the gas is oxidized to carbon dioxide with the liberation of iodine, the amount of iodine being directly proportional to the amount of CO gas present in the sample. The iodine thus liberated is titrated with a standard sodium thiosulphate solution, from which the concentration of CO may be computed. In the pyrotannic acid method the reaction depends on the introduction of CO-free blood into the sampling container in order to form CO-hemoglobin. Pyrotannic acid, when introduced after the blood has been in intimate contact with the gas for 15 min or longer, will give a characteristic carmine red color, the intensity of which will depend on the amount of CO-hemoglobin present. Still another technic for the detection of CO gas in air, and one which is very rapid and sensitive, is the thermo-electric method using a catalyst (18). The CO is oxidized to CO_2 in the presence of the catalyst with the liberation of heat. The quantity of heat which is liberated is measured directly on a millivoltmeter with the aid of thermocouples imbedded in the catalyst. Since the amount of heat liberated is directly proportional to the amount of CO gas present in the air, the millivoltmeter readings indicate the quantity of CO.

Benzol vapors may be detected in the air by the activated charcoal method, in which the vapors are absorbed by charcoal and the concentration determined by a computation from the increase in weight of the charcoal (19). The interferometer (20) has also been used for the detection of benzol in air, and if no interfering vapors are present, this device is very sensitive and yields rapid readings. However, neither the charcoal nor interferometer method is specific for benzol in the presence of other hydrocarbon vapors. The method developed by Smyth (21), which is a modification of a nitration method, has been found to be specific for benzol vapor in the presence of certain acetates and alcohols. In this method the air-vapor mixture is passed through an absorption device containing a mixture of equal volumes of sulphuric acid and fuming nitric acid. The benzene is converted to dinitrobenzene and then subjected to a chemical analysis in the laboratory.

Application of Test Methods

The various methods and instruments for the sampling and analysis of dusts, fumes, gases and vapors have been presented. A few examples of the practical application of the technic described in the study of the efficiency of industrial exhaust systems, may serve to indicate the value of such criteria in the present problem.

As a result of a prolonged study of the health of the workers engaged in the various occupations of granite cutting (1), it was possible to demonstrate that those persons engaged for many years in tasks associated with a dust exposure of less than 10 million particles per cubic foot of air were not suffering from silicosis or tuberculosis, the diseases most prevalent among these granite cutters. It was also possible to demonstrate that among these granite cutters the incidence of silicosis and tuberculosis, all other factors being equal, was directly proportional to the degree of dust exposure. The solution of the dust problem in the granite cutting industry therefore, resolved itself in the removal of the dust at its source, to an amount less than 10 million particles per cu ft, preferably by exhaust ventilation devices. Studies of the efficiency of such dust removal devices have been made using the dust determination at the worker's breathing zone as a final criterion (22). In Table 3 a comparison of air dustiness is presented between granite cutting plants using exhaust ventilation devices and those not equipped with such protection.

Table 3—Comparison of atmospheric dust conditions between two granite-cutting plants

Occupation	Average dust count in millions of particles per cubic foot of air; winter observations		
	Plants without efficifnt local exhaust system	Plants with efficient local exhaust system	
		Plant X	Plant Y
All pneumatic hand-tool operations	55.2	23.5	9.5
Surface cutting	45.0	15.3	10.6
Tool grinding	30.0	5.9	12.1
Sand blasting	6.9	3.5	5.5
General plant atmosphere	22.6	5.6	8.9

It is apparent from the results presented in Table 3 that in plant X the exhaust devices in use with pneumatic tool operations needed attention, since the dust concentrations associated with these operations were slightly higher than the prescribed standard. On the

other hand, the ventilation system in plant Y was apparently functioning satisfactorily at the time these studies were conducted.

In studying the degree of exhaust ventilation necessary to keep the dust at the worker's breathing level to an amount less than 10 million particles per cubic foot, the dust determination method again proved very useful (22). Fig. 4 presents the results of a study of the relation between the degree of air velocity at exhaust ducts and the amount of dust inhaled by granite cutters using various pneumatic tools. It is apparent from this figure that by maintaining an exhaust velocity of 1500 linear feet per minute at the face of the dust removal hood of the type investigated in the present study, the dust concentration at the worker's breathing zone will be less than 10 million particles per cubic foot.

The workers at the Harvard School of Public Health in conducting their studies on the removal of dust from granite cutting operations also used the dust determination technic as a final criterion of the efficiency of the ventilation system designed by them (23) (24). These investigators, as a result of their research, have been able to specify the correct design of local exhaust hoods and the minimum air flow requirements for the removal of dust generated by the use of pneumatic cutting tools in the granite cutting industry.

Mention has already been made of the use of the impinger apparatus in the determination of chromic acid mist in the air of chromium plating establishments (15). These determinations were made at chromium plating tanks which were provided with exhaust ventilation. The degree of exhaust ventilation was determined at the same time that samples of air were obtained for chromic acid analysis. Such a study revealed the effectiveness of the exhaust system in removing chromic acid mist from the air. In Fig. 5 the relation between the degree of air velocity at the exhaust ducts and the amount of chromic acid in the air is shown. It is evident from this study that in order to keep the chromic acid content in the air to an amount less than 1 milligram in 10 cubic meters (the minimum amount found to cause no damage to the nasal septum), at least 1500 feet per minute of air movement at the face of the duct is necessary, especially at the higher current densities commonly encountered in electroplating.

Again, in studying the efficiency of sandblast helmets used in the protection of men working inside sandblast rooms, it was found that a relationship existed between the amount of air supplied to the helmet and the concentration of dust inside the helmet during blasting. In an attempt to determine the optimum air volume to be supplied to such protective devices, it was necessary to obtain dust samples from inside the helmet while varying the air volume, at the same time maintaining the dust concentration in the sandblast room (outside the helmet) constant. Fig. 6 shows the results of such a study and clearly indicates that the positive supply of 6 cu ft of dust-free air per minute will protect a worker under the operating conditions now in practice in sandblast rooms. The ultimate criterion of protection, however, is the result of dust determinations of the air within the helmet, that is, the air actually breathed by the worker, and not the volume of air supplied (25).

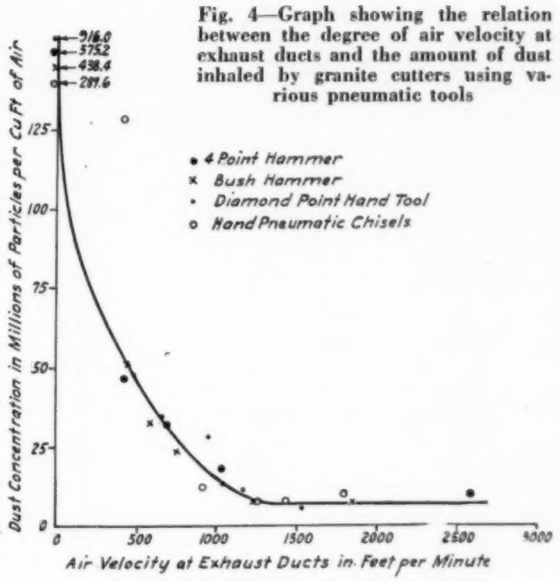

Fig. 4—Graph showing the relation between the degree of air velocity at exhaust ducts and the amount of dust inhaled by granite cutters using various pneumatic tools

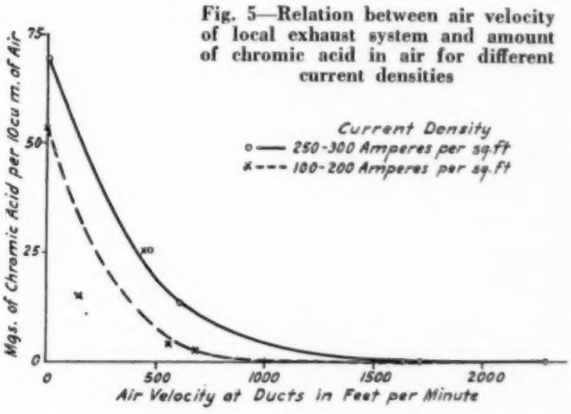

Fig. 5—Relation between air velocity of local exhaust system and amount of chromic acid in air for different current densities

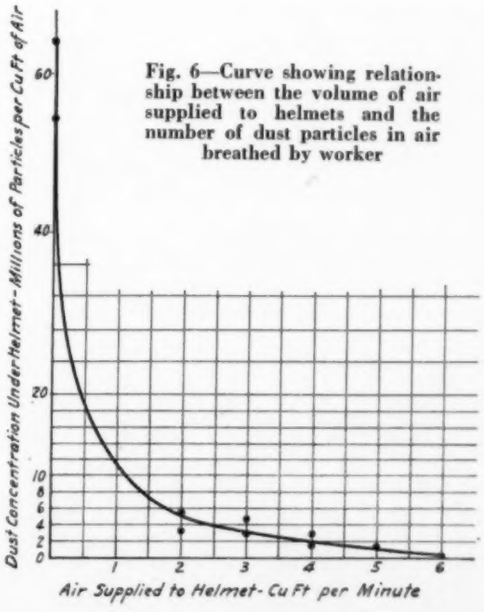

Fig. 6—Curve showing relationship between the volume of air supplied to helmets and the number of dust particles in air breathed by worker

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Local Chapter Reports

Cincinnati

April 10, 1934. Pres. H. N. Kitchell presided at the April meeting of the Cincinnati Chapter, which was held at the Engineers' Club, and welcomed the guests and members in his usual affable manner, inviting them to future meetings. He then introduced J. R. McConnell, Louisville, Ky., the speaker of the evening whose paper on Air Cleaning as Applied to Modern Air Conditioning was in reality a complete resumé of air cleaning for ventilation and industrial problems. The discussion was divided into three general sections: first, the sources, prevalence and malignancy of atmospheric dirt; second, the history, development and design of air cleaning devices; and third, the application of air cleaning equipment. The entire talk was illustrated with photos, sketches and charts.

Of particular interest in the first section of the discussion was a micro-photograph showing comparative sizes of normal blood corpuscle, human hair of average diameter, and dust particles of 2, 5 and 10 micron maximum dimensions. An accompanying chart showed the entire range of dust particle size and concentration based on micron and corresponding mesh designations and gave the application of various air cleaners to dust particles according to micron designation, placing the particles in their corresponding groups according to their natural movement in the air and marking the intermediate points, which divide visible from invisible dust, as well as indicating the particles encountered in the recent general discussions occasioned by industrial and health codes.

In the second section, the general development of air cleaners was traced through cloth screens, coke filled filters, surface filters, air washers, unit impingement filters, automatic impingement filters, surface filters, and special filters employing a combination of the foregoing designs. Of particular interest in this section was an explanation of how each basic design developed as the correlated change in dust type and concentration changed with industrial development and activity. Special attention was given to the development of filtering media in each group. In the wet or water units the development being not in media but in washer design; in the impingement, semi-dry or oil units, the development being in media including steel, glass, mineral wools, etc., and especially in the charging liquids used as viscous coatings for which exhaustive research is required; and in the dry or surface type cleaners the development of cellulose and other fiber media along with the application of cloth.

In the third division covering application, some interesting industrial applications were mentioned, including intake filters for engines and compressors, line filters, and range hood filters. Dust control and recovery in industrial applications were briefly covered and served as an introduction to the unique new dynamic precipitator which recently has been proven in industrial fields.

Discussion followed the questions of Messrs. Royer, Powers and Motz.

March 27, 1934. The Cincinnati Chapter held its March meeting at the Engineers' Club with 119 members and guests present.

This interesting open meeting was addressed by C. J. Kiefer, consulting engineer, on the subject of Distillery Engineering. Mr. Kiefer covered the principles of fermentation and distillation and the equipment used in distilleries and bottling plants. Sixty stereopticon slides were shown, illustrating in detail both obsolete and modern equipment and the general layout of the various departments of up-to-date plants.

A number of questions were asked to which Mr. Kiefer replied and said that experimenting was now going on to determine if whiskey could be aged in two years in heated warehouses, as well as in four years in unheated buildings. The temperature has much to do with the aging and consequently a difference has been noted between spring and fall made whiskey.

Cleveland

February 9, 1934. Pres. F. A. Kitchen called the gathering to order in the meeting room of the Cleveland Engineering Society with 25 members and guests present.

The treasurer reported a balance of $48 in the current account and said that the period for discounting of dues had elapsed.

The secretary, M. B. Wright, stated that an inquiry had been received from the Federal Screen and Weather Strip Mfg. Co. for a salesman to handle their product in the Cleveland territory.

The chairmen of the Membership Committees were asked to report. C. A. Wheeler of the Blue Team proposed the name of F. A. Rodgers for membership in the Cleveland Chapter, election to become effective at the time of Mr. Rodgers' election to the national Society. This nomination was seconded and carried. Philip Cohen of the Red Team nominated F. R. Dickinson for membership in the Cleveland Chapter to become effective at the time of his election to the national Society. This nomination was also seconded and carried.

Secretary Wright read the report of the Committee on Cleveland Slum Elimination and Mr. Cohen moved that the report

of the committee be accepted. No action was taken on the recommendations of the committee.

C. R. Sabin of the Cleveland Engineering Society discussed the State Engineers License Law.

This was followed by President Kitchen's report of the Annual Meeting of the A.S.H.V.E.

John Howatt, Chicago, Ill., first vice-president and chairman of the Committee on Research of the A.S.H.V.E., was introduced by President Kitchen as the evening's speaker.

The subject of Mr. Howatt's talk was the Selection and Maintenance of Mechanical Equipment for School Buildings. He discussed in detail such factors as conditions peculiar to the locality, low vs. high pressure steam, size of plant, the use of steam in the building, useful life of the equipment, and fuels. At the conclusion of his talk, Mr. Howatt answered several questions asked by members.

Before the meeting was adjourned, a rising vote of thanks was extended to Mr. Howatt.

Illinois

April 9, 1934. The Illinois Chapter held its regular meeting at the Hotel Sherman with an attendance of 63. Pres. C. W. DeLand presided and the minutes of the previous meeting were read and approved.

Secy. J. J. Hayes read a letter from A. C. Willard, newly elected president of the University of Illinois, received in reply to a telegram of congratulations from the Illinois Chapter.

A communication was also read from J. H. Hucker, chairman of the Publicity Committee, Philadelphia Chapter, outlining the program of the Semi-Annual Meeting to be held at Buck Hill Falls, Pa.

Three new members were elected into the Illinois Chapter: E. B. Evleth, G. G. Turner and J. T. Machen.

President DeLand then introduced P. L. Davidson, Philadelphia, Pa., who described an air conditioning installation recently made in a combination office, banking and retail store building of the Philadelphia Savings Fund Society. The fact that the entire building is air conditioned has proved to be one of the largest factors in persuading tenants to lease space, in spite of a rental comparable with that of buildings more advantageously located.

Mr. Davidson emphasized the necessity of close cooperation between the architect and the air conditioning engineer, the exercise of great ingenuity by both, and a mutual respect for each other's problems.

Research on new methods of air distribution was being carried out in laboratories at the time the air conditioning of the upper floors was being discussed. These methods, incorporated into this building without field test, consist of introducing air into the conditioned spaces at high velocity and pressure, 35 deg below room temperature, with a duct system so designed that it cannot become unbalanced by a tenant opening his window.

Mr. Davidson described in detail the construction of the entire system and the individual units of apparatus employed. Numerous slides illustrated much of this equipment, together with typical air conditioning units. The speaker concluded his talk with a brief outline of the present state of the art of air conditioning and predicted a few of the problems of the future.

A lively discussion followed, entered into by numerous members and guests. An enthusiastic vote of thanks testified to the interest aroused by this paper.

Massachusetts

March 12, 1934. The March meeting of the Massachusetts Chapter was held at the Harvard School of Public Health with an attendance of 60 members and guests.

Pres. Leslie Clough called the meeting to order and stated that the new By-Laws were adopted at a special meeting on February 26, subject to approval by the Council of the Society. In accordance with the new regulations, he announced the nominees for election by ballot to the Board of Governors as follows: J. W. Brinton and A. B. Jackson for one year; C. T. Flint for two years; and G. B. Gerrish, W. A. McPherson, and Edgar Shaw for three years.

Philip Drinker, Harvard School of Public Health, then introduced W. W. Wells, a member of the staff, who showed and operated a machine which he had fabricated for collecting germ samples. He explained that by counting under a powerful microscope the deposit of known germ life from a specified enclosure on a small slide, he could estimate closely the germ content of the air in the enclosure. Mr. Wells said that his experiments were in their infancy but he saw opportunities in his mechanism for worthwhile research.

Mr. Drinker followed with an address on the application of commercial or other filters to Mr. Wells' machine. He showed microscopic representations of rag weed and other pollens on a screen and injected a humorous angle by showing a slide containing dust from his home. He admitted his inability to identify all of the particles on the slide, but pointed out coal ash and airedale hairs and assured the audience that his house was in order and clean on all occasions.

Both addresses were brief but very interesting. Refreshments were served by Professors Philip Drinker and C. P. Yaglou, hosts to the Chapter.

Michigan

March 19, 1934. About 95 members and guests of the Michigan Chapter met at the Wardell Hotel for dinner and a meeting, at which Pres. H. E. Paetz presided and read a notice of the appointment of A. C. Willard to the presidency of the University of Illinois.

After roll call, R. K. Milward reported $208.00 in the treasury. Six applications for limited chapter membership were read.

President Paetz announced that the following members would serve on the Nominating Committee: L. L. McConachie, chairman, E. E. Dubry, and J. F. McIntire.

Mr. Milward was appointed chairman of a committee to arrange for a golf meet in May, with F. J. Feely as vice chairman.

Recognition was given to the services of Messrs. Armstrong and Fuller in arranging for the speakers of the evening.

President Paetz introduced Harvey Campbell, of the Detroit Board of Commerce, who acted as toastmaster.

The Chapter was honored by the presence of C. V. Haynes, president of the A. S. H. V. E., who gave a few comments regarding the operation of the Society.

Arthur Herske gave a talk on the condition of the construction industry and what the future holds. He stressed the idea that considerable business is to be had by improvements to present buildings.

Mr. Campbell presented Malcomb Bingay, of the *Detroit Free Press,* who gave an inspiring talk on Our Detroit and received much applause.

The meeting adjourned at 10:00 p. m., according to the report of the secretary, Tom Brown.

Ontario

April 2, 1934. Dinner was served at 6:00 p. m. to 64 members and guests of the Ontario Chapter, who met at the Royal York Hotel for the regular meeting.

Pres. W. P. Boddington read a report of the Nominating Committee consisting of W. G. F. Sheppard, *chairman,* A. S. Leitch and E. T. Flanagan. The following names were submitted for offices in the Ontario Chapter for the ensuing year 1934-1935:

President—W. R. Blackhall.

Vice-President—M. W. Shears.

Secretary-Treasurer—H. R. Roth.

Board of Governors—J. S. Paterson, G. A. Playfair, Thomas McDonald, and W. P. Boddington.

President Boddington asked that M. F. Thomas and H. D.

Henion act as auditors of the treasurer's books and make a report at the annual meeting in May.

The following committee was announced by President Boddington to be in charge of the May meeting: H. H. Angus, *chairman,* H. B. Jenney, Thomas McDonald, and J. S. Wood.

After the business had been completed, President Boddington called upon the speaker, E. L. Hogan, Detroit, Mich., who gave an instructive and interesting talk on Air Conditioning.

Following the address, the meeting was open for discussion at which time many points on the application of air conditioning equipment were discussed.

Pittsburgh

March 12, 1934. Members and guests of the Pittsburgh Chapter met in the auditorium of the U. S. Bureau of Mines for the regular March meeting, with Pres. P. A. Edwards presiding.

R. B. Stanger, chairman of the Program-Publicity Committee, stated that plans were progressing for an interesting and instructive meeting in April. The Membership Committee reported through its chairman, T. F. Rockwell, that it was actively at work.

President Edwards appealed to the members to pay their current and delinquent dues.

A telegram from the Philadelphia Chapter inviting the Pittsburgh Chapter to attend the Semi-Annual Meeting in the Pocono Mountains was read and E. C. Evans talked about the beauty of the location chosen and urged that all Pittsburgh members make an effort to go.

After reading and discussing Policies A and B written by the Pittsburgh Chapter of the *American Institute of Architects* concerning the employment of CWA professional help, it was moved by C. M. Humphreys, seconded by R. B. Stanger, and carried that both policies be subscribed to by the Pittsburgh Chapter of the A. S. H. V. E.

The new plans of the Society's Council concerning the payment of delinquent dues was discussed by Messrs. McIntosh and Evans.

E. H. Riesmeyer, Jr. presented the Chapter's guest and speaker, W. E. Stark, Cleveland, who is a member of the Council and of a number of Society committees. Mr. Stark spoke on Winter Air Conditioning for the Residence and traced its evolution. By quoting from old text books on the subject, he clearly showed how far the engineer has progressed in the heating field in 30 years. By drawing upon data obtained as a result of his research work, Mr. Stark described the requirements of present day winter heating, outlined the principles of what he considered to be an ideal winter conditioning plant, and imagined what the future might bring.

After a long and interesting discussion, the meeting was adjourned at 10:15 p. m.

St. Louis

April 4, 1934. Pres. Paul Sodemann called the members and guests of the St. Louis Chapter to order in the Kingsway Hotel, and outlined the program for the Semi-Annual Meeting of the Society and the invitation from the Philadelphia Chapter to members of the St. Louis Chapter.

Secy. A. L. Walters read a letter from A. V. Hutchinson in regard to the election of A. C. Willard as president of the University of Illinois, also a copy of a letter of congratulation sent by L. W. Moon on behalf of the Chapter and Professor Willard's reply.

C. R. Davis, chairman of the Entertainment Committee, outlined the plans for the party at Joe Garivelli's on April 14.

The chairman of the Membership Committee, D. J. Fagin, presented the application of L. L. Benoist, Mt. Vernon, Ill.

It was announced that C. A. Pickett had talked on air conditioning before the St. Louis Electrical Board of Trade.

W. P. Hacker was the speaker of the evening, his subject being Experiences of an Engineer in China. His talk was very interesting and was greatly enjoyed by those present. Mr. Hacker was given a rising vote of thanks, after which the meeting was adjourned.

Western New York

April 9, 1934. A very interesting meeting of the Western New York Chapter was held at the Touraine Hotel and was conducted by First Vice-Pres. W. F. Johnson in the absence of Pres. J. J. Yager. The attendance of 19 for dinner was increased to 31 for the meeting.

L. A. Harding was the speaker and gave an enlightening, as well as historic lecture, on Refrigeration—Past and Present. A lively discussion followed with a number of questions being asked of Mr. Harding and also of Messrs. Mullenburg and Crique.

A motion was made and seconded that Secy. P. S. Hedley write A. C. Willard, congratulating him on his election to the presidency of the University of Illinois.

A second motion was made and seconded for the secretary to write a letter of sympathy to Mrs. Anderson, expressing the sincere regret of all members for the death of their good friend, F. Paul Anderson, dean of engineering of the University of Kentucky.

Mr. Johnson suggested that an effort be made to increase the attendance at the May meeting and appointed a temporary committee of D. J. Mahoney, *chairman,* Joseph Davis, and M. C. Beman, to assist in this work.

A rising vote of thanks to Mr. Harding, who was the first president of the Chapter and a past president of the A. S. H. V. E., closed a very instructive and enjoyable meeting.

March 12, 1934. The first business meeting with Pres. J. J. Yager presiding was held at the Touraine Hotel with an attendance of 20 at dinner and 32 at the meeting.

The speaker, F. I. Raymond, of Chicago, was introduced by D. J. Mahoney. The subject, Principles of Steam Conservation, was very ably covered by Mr. Raymond and considerable interest was evidenced by the number of questions asked after his talk.

Roswell Farnham reported that approximately 25 members from the Western New York Chapter attended the 40th Annual Meeting of the Society in New York and the Heating and Ventilating Exposition.

A rising vote of thanks to the speaker closed the meeting.

Paul McDermott with J. M. Laboratory

Paul McDermott

Paul F. McDermott completed nearly 5 years' service as research engineer at the A. S. H. V. E. Research Laboratory in Pittsburgh, Pa., before he joined the research staff of Johns-Manville Co., Manville, N. J., where he is now employed in the Materials Development Section as physicist.

While at the Research Laboratory in Pittsburgh, Mr. McDermott successfully completed many studies of importance in the heating and ventilating field, the problems involving the analysis of film resistance coefficients, wind velocity and temperature gradients away from a surface, sun effect, cold walls and their relation to the feeling of warmth, ventilation requirements of garages, the effectiveness of window shading appurtenances, and heat transmission. Mr. McDermott came to the A. S. H. V. E. Research Laboratory in July, 1929, after his graduation from the University of Washington where he had majored in physics. As a member of the Society's Laboratory staff under Director Houghten he was author of, or worked on, the preparation of many technical papers and reports which were presented at meetings of the Society.

New Code Authority Membership Announced

In a recent announcement of the National Recovery Administration the method of selection of personnel of several code authorities is given. Two of the industries are of special interest to Society members.

Non-Ferrous and Steel Convectors Manufacturing Industry: Reuben N. Trane, president, The Trane Co., LaCrosse, Wis.; M. S. Dillman, manager, Concealed Radiator Dept., American Radiator Co., New York, N. Y.; A. G. Dixon, manager, Heating Division, Modine Manufacturing Co., Racine, Wis.; and L. O. Mandell, manager, Heating Division, C. A. Dunham Co., Chicago, Ill.

Unit Heater and/or Unit Ventilator Manufacturing Industry: J. I. Lyle, president, Carrier Engineering Corp., Newark, N. J.; G. E. Otis, vice-president, The Herman Nelson Corp., Moline, Ill.; J. F. G. Miller, vice-president, B. F. Sturtevant Co., Hyde Park, Mass.; and R. J. Tenkonohy, Airtherm Manufacturing Co., St. Louis, Mo.

DeNeille Elected President of Heating Contractors

The 45th annual convention of the *Heating, Piping and Air Conditioning Contractors National Association* was held at the Ambassador Hotel, Atlantic City, N. J., May 22-25, and the program was featured by several addresses on The National Recovery Program, The Construction Code and Its Administration, and reports of the officers and standing committees.

An address on Radiators and Convectors was delivered by Prof. A. P. Kratz of the University of Illinois and Willis H. Carrier, Newark, N. J., spoke on Air Conditioning in 1934.

A brief word of greeting was delivered by C. V. Haynes, president of the A. S. H. V. E., and the merchandising session included addresses on Modernization and Merchandising Automatic Heating Equipment.

At the final session the officers and directors were elected and for the coming year the following will guide the association's affairs: *President*—J. Lawrence DeNeille, St. Louis, Mo.; *vice-president*—John H. Zink, Baltimore, Md.; *treasurer*—M. J. Chagnard, Louisville, Ky.; *secretary*—Joseph C. Fitts, New York, N. Y.; *secretary emeritus*—Henry B. Gombers, East Orange, N. J.; *educational director*—S. Lewis Land, New York, N. Y.; *Board of Directors*—J. L. DeNeille, chairman, St. Louis, Mo., Maurice J. Chagnard, Louisville, Ky., George H. Dickerson, Chicago, Ill., Curtiss N. Gilley, San Francisco, Calif., Justin A. Kiesling, Houston, Tex., Walter Klie, Cleveland, O., J. J. Nolan, Memphis, Tenn., Thomas O'Callaghan, Boston, Mass., H. Merwin Porter, Minneapolis, Minn., Ray L. Spitzley, Detroit, Mich., George J. Stoehr, Milwaukee, Wis., E. W. Tompkins, Albany, N. Y., Robert D. Williams, New York, N. Y., and John H. Zink, Baltimore Md.

A dinner dance and entertainment at the Ambassador Hotel was the main social feature and during the meeting the ladies were entertained at tea and the men played a Kicker's Handicap match on the course at Northfield.

Furnace Manufacturers Mid-year Meeting

The mid-year meeting of the *National Warm Air Heating and Air Conditioning Association* was held at the Stevens Hotel, Chicago, June 12-14. The first day was devoted to a meeting of the Board of Directors and various committees and Pres. H. T. Richardson opened the first session, June 13, which was devoted to business discussions by leaders in the industry and a report on codes by Prof. J. D. Hoffman.

In the afternoon session a combined meeting of manufacturers and dealers was held and the principal topic of discussion was the N. R. A. Codes of Our Industry.

On Thursday morning Research was the subject of discussion and F. G. Sedgwick, chairman of the Research Advisory Committee, presided. Arthur C. Willard, president-elect of the University of Illinois, addressed the meeting on Fifteen Years of Research in the Furnace Heating Industry, after which Edwin A. Scott, editor of *Sheet Metal Worker*, discussed the question of Tabulating and Publishing Research Data in Book Form.

The Insulation of Homes was the subject of a paper by Prof. A. P. Kratz, University of Illinois, and a progress report on warm air heating research involving tests on thermostat location, registers, furnaces and air washers was presented by S. Konzo of the research staff. Business opportunities and reports of committees featured the afternoon session.

Victor N. Welamb Dies

Victor N. Welamb, president of the V. N. Welamb Co., Philadelphia, Pa., died May 16, 1934, in the Eaglesville Sanitarium, after a short illness. He was born in Trelleborg, Sweden, January 12, 1878, and graduated from the Hanover University, Hanover, Germany.

For many years he was connected with Baker, Smith & Co., New York, and then was placed in charge of their Philadelphia office. In 1922 the firm of Muench, Welamb & Co., heating and plumbing contractors, was organized and when this partnership was dissolved in 1924 the V. N. Welamb Co. was formed. This organization was highly successful from its beginning and installed heating and ventilating equipment in many of the outstanding buildings erected in the Philadelphia area in recent years. This list includes the Penn Athletic Club, Gimbel Bros. Store, Strawbridge & Clothier Store, Hahnemann Hospital, Bryn Mawr

Mr. and Mrs. Ray Spitzley, Mr. and Mrs. H. M. Hart

A. V. Hutchinson, A. P. Kratz, W. H. Carrier, C. V. Haynes

John S. Jung, H. N. Booth, K. L. Ziesse, O. J. Prentice

On the Atlantic City Boardwalk during the Annual Convention of H. P. & A. C. C. N. A.

Hospital and the Irvine Auditorium, University of Pennsylvania.

Mr. Welamb was regarded by his contemporaries as an authority on the ventilation of large buildings. His kindness and his genial disposition endeared him to a host of personal and business friends. The AMERICAN SOCIETY OF HEATING & VENTILATING ENGINEERS and the Philadelphia Chapter in particular have suffered the loss of a loyal member of sterling character. Mr. Welamb was a Mason and also a member of the *American Society of Swedish Engineers*, the Penn Athletic Club and the Bankers & Manufacturers Club. He is survived by his widow, Cordelia A. Welamb, to whom the Officers and Council of the Society extend their sincere sympathy.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 12 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by July 16, 1934, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

Candidates	*Proposers*	*Seconders*
ANDREWS, GEORGE H., Partner, Frank P. Andrews & Son, Pittsburgh, Pa.	G. S. McEllroy F. C. McIntosh	P. A. Edwards F. A. Gunther
CHARLES, THOMAS J., Pres., Metropolitan Air Cond. Corp., New York, N. Y.	H. W. Fiedler H. L. Alt	H. G. Meinke A. F. Hinrichsen
DICKENSON, FREDERICK R., Dist. Mgr., American Blower Corp., Cleveland, Ohio.	Philip Cohen F. A. Kitchen	W. E. Stark M. B. Wright
JEPERTINGER, RICHARD C., Treas. & Chief Engr., Air Cond. Equip. Corp., Milwaukee, Wis.	C. H. Randolph Ernest Szekely	J. H. Volk E. A. Jones
KYLBERG, V. C., Htg. & Refrigeration Engr. & Contr., Maplewood, N. J.	W. A. Evans G. G. Schmidt (*Non-Member*)	Kennedy Duff M. W. Ehrlich
PLUM, LEROY H., Industrial Engr., Minneapolis-Honeywell Regulator Co., Philadelphia, Pa.	C. G. Binder K. W. Rohlin	F. D. Mensing E. N. Sanbern
SCHICK, KARL W., Sales Engr., Minneapolis-Honeywell Regulator Co., Cleveland, Ohio.	Philip Cohen F. A. Kitchen	W. E. Stark M. B. Wright
SPENCE, M. R., Asst. Purch. Agt., Rundle & Spence Mfg. Co., Milwaukee, Wis.	V. A. Berghoefer A. M. Wagner	C. W. Miller C. H. Randolph
STEWART, J. C., Owner, Acme Plumbing Shop, Regina, Sask., Canada.	D. F. Michie J. H. Leonard	William Glass F. R. Agnew (*Non-Member*)
STROUSE, S. W., Sales Engr., Cooney Refrigeration Co., Inc., Buffalo, N. Y.	W. H. Carrier M. C. Beman	B. C. Candee Joseph Davis
WALTON, C. W., JR., Air Cond. Engr., Rockefeller Center, Inc., New York, N. Y.	A. C. Buensod A. V. Hutchinson	E. T. Lyle
WEITZEL, PAUL H., Student, Lehigh University, Bethlehem, Pa.	A. E. Kriebel H. H. Erickson	W. R. Eichberg C. A. Rhea

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

HAYTER, BRUCE, Chief Engr., Inst. of Thermal Research, American Radiator Co., Yonkers, N. Y.

SHENK, D. H., Assoc. Prof. of Mech. Engrg., Clemson Agricultural College, Clemson College, S. C.

STEWART, C. W., Engr., Ilg Electric Vtg. Co., New York, N. Y.

ASSOCIATES

MORSE, F. W., Asst. Gen. Sales Mgr., Chamberlin Metal Weather Strip Co., New York, N. Y.

STEGGALL, HOWARD B., Br. Mgr., U. S. Radiator Corp., Pittsburgh, Pa.

TOBIN, J. F., Sales Engr., American Blower Corp., Chicago. Ill.

JUNIORS

FITZSIMONS, J. P., Engr. and Estimator, Robt. Fitzsimons Co., Ltd., Hamilton, Ont., Canada (*Advancement*).

HOLMES, RICHARD E., Air Cond. Engr., Westinghouse Elec. & Mfg. Co., Springfield, Mass.

REED, I. G., Asst. Supt. and Chief Engr., Grant Bldg., Inc., Pittsburgh, Pa.

STUDENT

GAULT, G. W., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.